AUTOMOBILE ENGINEER

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Design, Materials, Production Methods, and Works Equipment

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Concerted Effort

FREE interchange of thought and the expression of ideas between people is generally accepted as being good for the community in which they live. Where, for instance, would medicine and science stand to-day had Pasteur, Lister, Otto, Curie, and a myriad others, kept their ideas to themselves instead of sharing their invaluable discoveries. Similarly, the unfettered exposition of technical design and development, coupled with constructive criticism thereof, must necessarily speed the evolution of the product, to the ultimate good of both maker and user. Such co-operative effort is one of the signs of an enlightened society, and even those who do not always practise it enthusiastically, at least pay lip service to the principle.

Learned Societies

Because they materially stimulate and encourage such action, the learned societies of any land are a leading and invaluable factor in co-operative effort. Neither men nor their organizations, however, can live for themselves alone, and these societies depend for their strength and influence upon the support provided by their members. They are not money-making concerns, and their membership depends upon the number of interested persons qualified to join them. Entry into their ranks becomes progressively harder as industries develop and higher standards of professional education and knowledge are demanded in their technicians. No one would dispute the fact that this is perfectly right and proper.

The British automobile industry ranks as the third largest in the land and its products are now sold in considerable numbers in most places on the earth's surface. The maintenance of this position is, however, by no means inevitable and assured. Increasing competition from various formidable sources is early to be expected. Everything will depend upon the ability of our technicians to keep ahead in contemporary design and development.

What opportunities have they to share and increase their knowledge through discussion with their fellows? For this, they must in the main look to our own particular learned society, namely, the Automobile Division of the Institution of Mechanical Engineers, formerly the Institution of Automobile Engineers. Many lament the end of the separate existence of that Institution which undoubtedly played a valuable part in the development of British automobile engineering. There seems little doubt, however, that the industry did not in return give it the support and encouragement it merited.

It appeared to resent the fact that it could not possess it financially and therefore withdrew the moral support to which such an Institution was justly entitled. It is quite extraordinary that the Government and armed services should have refused to accept its membership as a technical qualification. Contrast this situation with the manner in which the Government of the United States turned to the Society of Automotive Engineers for assistance when war threatened. A glance at the proceedings of that society reveals the wide range of its activities and shows that it is indeed a power in its land, and beyond. Most automobile engineers in this country make it their business to study the papers and technical work of this great society.

Membership

Bearing in mind the magnitude of the automobile industry in Great Britain, it is surprising that at the beginning of 1950 there were only 3,000 corporate members and some 600 graduate and student members of the Automobile Division of the Institution. While the number of skilled technicians is admittedly small in relation to the total number of people employed in any industry, these figures are disturbing, especially with regard to the younger members. What is the reason? It cannot be one of expense, as the subscriptions to our technical institutions are very low. It seems hard to believe that it can be lack of interest. Not only does membership of a professional society yield dividends in the form of knowledge and prestige, but also invaluable personal friend-

ships based upon common interests. These, we know from experience, can be sound and enduring over very many years.

Now if there is any single factor likely to influence quite strongly the growth of usefulness of any engineering institution, it is the employers' attitude to employee participation. If those in high office in our great manufacturing concerns are well disposed towards the outside technical activities of their employees, then these staffs will work enthusiastically for, and with, the appropriate technical organizations. If, on the other hand, encouragement is but lukewarm, or worse still there is open or overt discouragement, then the worst effects may be anticipated. Unfortunately, such a very shortsighted policy is in fact being evidenced in some quarters.

Not only do certain employers look on these outside technical activities with disfavour, but in some instances actually pursue this narrow and ill-conceived policy to the point of raising obstacles. If British automobile engineering is to maintain its front rank position, its engineers and designers must take their rightful place in the corporate life of the Automobile Division of the Institution of Mechanical Engineers. Only in this way will this organization fulfil its function now, and also be built for the future as a training and proving ground for the technical men on whom the industry must at all times depend. Nothing could be more shortsighted than the view that there can be one-way traffic in science or engineering. Such traffic cannot go on indefinitely, there must be a return. Any individual who is to derive the fullest benefit from membership of a learned society must himself serve that society in return, if it is to prosper as it should.

While a paid staff takes care of all the routine work, the actual running of such a society is carried out by the voluntary work of its members who serve on the Council and various Committees. Much of that work is necessarily done in working hours which may mean the use of as much as a full working day each calendar month, when travelling time is taken into account. Even so, that is not too great a price for industry to pay for the increased knowledge and keenness which its technicians acquire through playing their part in the wider life of the engineering community.

Speeding the wheels of production is doubtless of prime importance, but checking employees' technical activity and development is no way to achieve this. Let employers rather actively encourage both senior and junior engineers to qualify and take their place in building up a fund of technical knowledge which will render our industry enduring in its lead and make for progress through the years, so that we retain our reputation as producers of the finest cars and heavy vehicles in the world.

Mechanical Handling

IN almost every industry material handling charges are an important factor in total costs. There are signs that this is being increasingly recognized throughout British industry, but there are still very wide differences between the well-equipped and the poorly equipped British factories. In general, the level throughout industry in this country is much lower than it should be. There is still far too much use of human effort and too little use of mechanical equipment for handling materials.

In some organizations the failure to make use of mechanical methods is no doubt due to lack of information regarding the wide range of equipment that is now available. In others, although mechanical handling equipment is installed, it is not used to the best advantage. The Mechanical Handling Exhibition and Convention, sponsored by our associated journal, *Mechanical Handling*, to be held in the Grand and National Halls, Olympia, from 6th to 17th June, 1950, should be of great interest and value to the engineers of such organizations.

It will also be of value to the engineers of well-equipped factories, since it will afford an opportunity of examining and comparing the most up-to-date British products for practically every type of material handling.

In addition to the very wide range of exhibits, there will also be a convention at which Papers on many different aspects of material handling will be delivered. Of these, two would appear to be of particular interest to production engineers in the automobile industry. These are *Material Handling—a Job of Production Engineering* to be delivered by W. Puckey, M.I.P.E., at 3 p.m. on Wednesday, 7th June, and *Application of Overhead Chain Conveyors to Factory Problems*, by K. D. Warwick, A.M.I.Mech.E., at 11 a.m. on Thursday, June 8th.

It is generally agreed that material handling problems have received much more attention in the U.S.A. than in this country, and the Convention will afford an opportunity for deriving information about present-day American practice. On Monday, June 12th, at 3 p.m., members of the specialist team on mechanical aids, recently returned from the U.S.A., will discuss their report and will reply to questions.

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THE BUICK "SPECIAL"

The Latest Model, fitted with Dynaflow Hydraulic Torque Converter

THE products of the Buick Motor Division of the General Motors Corporation occupy an intermediate position in the group's range, which extends from the Chevrolet, in which low selling price is the main consideration, to the Cadillac, some models of which represent the highest level in luxurious performance.

The Buick tradition is, if anything, conservative, and the vehicle embodies, in the main, design principles that have been followed by the organization for many years. The coil spring and wishbone suspension for example, in every detail of its design, exemplifies what appears to be the universal practice of the American motor industry, which evidences a complete uniformity so noticeably absent among English designers.

The rear suspension, however, which is by coil springs and a torque-tube back axle with cross-stabilizer link, is a practice originated by Buick many years ago, but not employed by other American concerns. The straight-eight engine is orthodox and straightforward while the cross-braced frame is simple and practical, and fulfils its purpose without an excessive amount of welding and "boxing."

GENERAL SPECIFICATION

ENGINE. Straight-eight, 3-3/32in bore, 4⅛in stroke, capacity 258 cu in (4·067 litres), Overhead valves, Zero-Lash tappets, compression-ratio 7·2 to 1 (premium fuel advised). Maximum B.H.P. 122 at 3,600 r.p.m. Maximum torque 216 lb feet at 2,000 r.p.m.

TRANSMISSION. Dynaflow torque converter with two-speed and reverse hydraulically controlled transmission. Converter torque multiplication (stalled) is 2·25 to 1. Emergency low gear and reverse both 1·82 to 1. Maximum overall torque multiplication 4·1 to 1.

BACK AXLE. Hypoid in banjo casing, torque tube and Panhard rod, coil spring suspension. Ratio 3·9 to 1.

FRONT SUSPENSION. Wishbone and coil spring. Stabilizer bar.

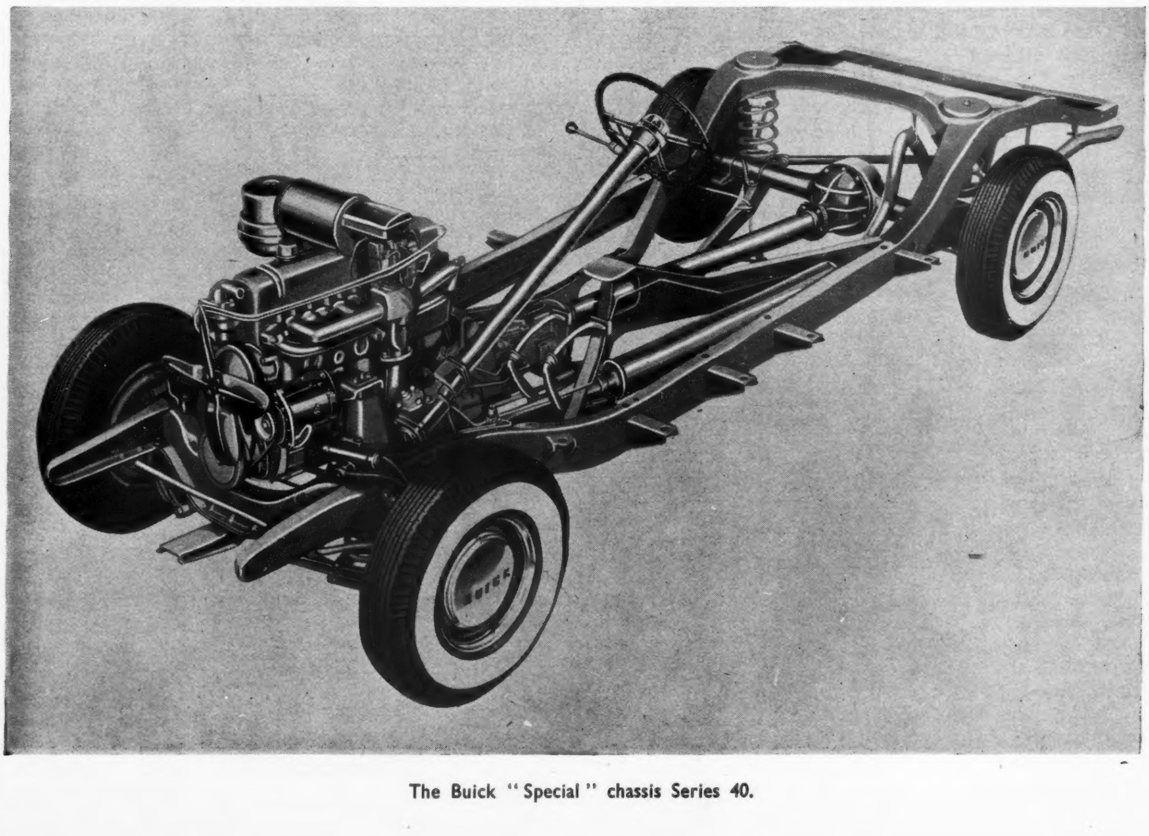
DIRECT-CONNECTED STEERING. Saginaw ball-bearing steering box. Box ratio 19·8 to 1. 4·25 turns lock to lock.

BRAKES. Bendix Duo-Servo, hydraulically operated, 12in drums, shoes 1¾in wide, 53 per cent front and 47 per cent rear braking. Foot-operated parking brake with push-button release.

The Buick "Special," which is the cheapest model produced by the firm, is, however, remarkable for offering at an extra charge of 212 dollars, the Dynaflow hydraulic torque converter previously only obtainable on the larger models.

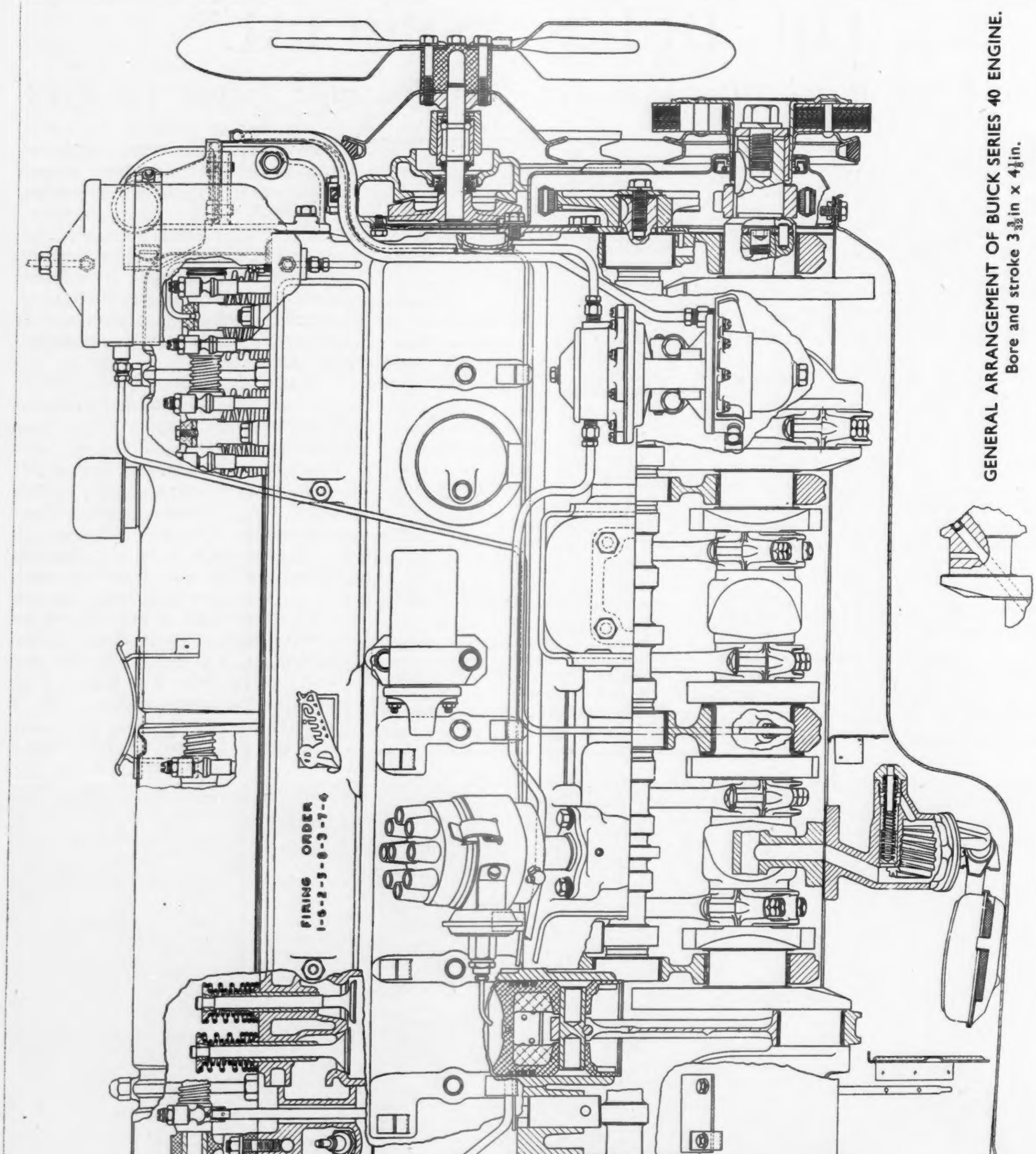
When the three-speed synchromesh gearbox is fitted, the engine has a compression-ratio of 6·6 to 1, gives a maximum b.h.p. of 115 at 3,600 r.p.m. and a maximum torque of 212 lb ft at 2,000 r.p.m., these figures being for the bare engine without accessories. The axle-ratio is 4·1 to 1, the tyres being 7·60—15, fitted on rims ½in wider than American standard. The six-seater four-door car has a kerb weight of 3,900 lb and the power-weight ratio is obviously excellent, as is evidenced by the fact that the first speed is as high as 2·67 to 1, second speed having a ratio of 1·66 to 1.

When the Dynaflow converter is

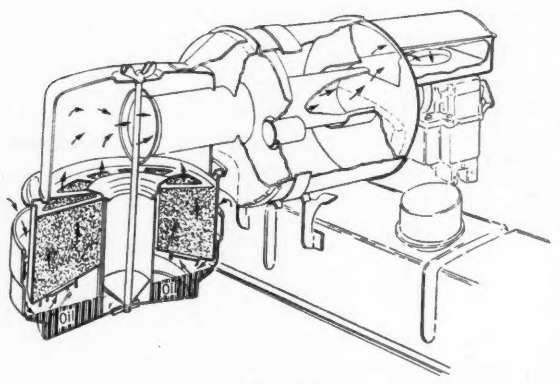


The Buick "Special" chassis Series 40.

THE BUICK "SPECIAL" (*Continued*)



GENERAL ARRANGEMENT OF BUICK SERIES 40 ENGINE.
Bore and stroke $3\frac{3}{32}$in x $4\frac{1}{8}$in.



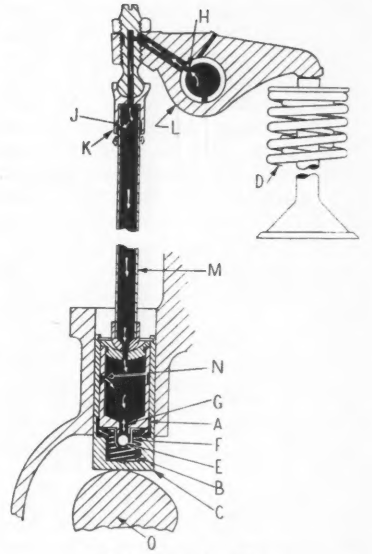
Air filter and silencer.

fitted several other changes are also made. The axle ratio is raised to 3·9 to 1 while the compression-ratio goes up to 7·2 to 1 and maximum torque to 216 lb ft. The engine is also fitted with hydraulic tappet adjustment. The object is to render the power unit as unobtrusive as possible. A slight "fussiness" might otherwise result from the fact that engine speed, with the hydraulic converter, changes more rapidly with acceleration than with direct drive and is, in general, considerably higher when climbing a gradient or picking up speed.

At the same time, the fact that the converter is continually varying the ratio according to load means that the engine is never called upon to develop high torque at low speeds. A higher compression-ratio can therefore be employed to advantage without risk of "pinking."

A speed of 90 miles per hour is stated to be obtainable with either model, and, in spite of the lower test-bed efficiency of the converter, as compared with the orthodox transmission, the fuel consumption is almost identical. In certain circumstances, especially when running at reduced speed for long periods, a good deal of heat is generated by hydraulic losses in the converter. An oil cooler is therefore fitted, taking its cooling water from the main system, the radiator being enlarged by some 18 per cent when the torque converter is fitted.

The torque converter, with its associated emergency low and reverse transmission, is of exceptional interest. It is particularly noticeable that extreme care has been taken in the design to provide for every contingency. It is remarkable how the exceedingly complicated hydraulic control mechanism has been reduced to a practical production job. What might well have been a maze of cross-connected pipes and drillings has been replaced by channels die cast in diaphragms closed by gaskets and cover plates. Not only is production cost vastly reduced, but all the passages are open to inspection on assembly and the absolute cleanliness so essential in piston-valve controlled hydraulic mechanisms is quite easy to secure.



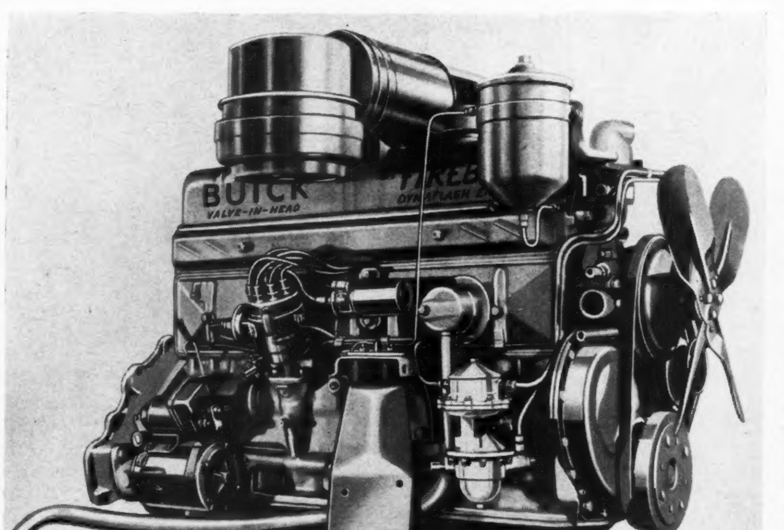
Hydraulic valve lifter.

A. Plunger. B. Spring. C. Body. D. Valve springs. E. Ball. F. Ball retainer. G. Feed hole. H. Feed passage. J. Bleed hole. K. Shroud. L. Rocker arm. M. Push rod. N. Oil groove and return hole. O. Camshaft.

Engine

The straight-eight overhead-valve engine has a bore of $3\frac{3}{32}$in and a stroke of $4\frac{1}{8}$in, so that the piston displacement is 248·1 cubic inches (4·067 litres). The block is cast integral with the crankcase, the sump joint face being $2\frac{3}{8}$in below crankshaft centreline. The crankshaft layout is orthodox, and can be described as a four-cylinder shaft with two-cylinder shafts added at each end at right angles.

There are five main bearings and the cylinders are spaced in pairs, there being no water between the barrels of each pair but ample cooling between the pairs. The jackets do not extend to the full length of the barrels but only just beyond the piston stroke. The tappet guides are cast integral and a long pressed-steel cover encloses the push-rods, the upper edge of this cover taking a bearing on the cylinder head, instead of on the block, a considerable simplification of the main casting being the result. A sandwich plate takes the timing case, and the bell-housing is separate, having an inturned flange which reduces the overall dimensions of the block appreciably.



Buick Series 40 engine.

Crankshaft and connecting rods

Having integral counterweights, the

THE BUICK "SPECIAL" ***(Continued)***

shaft runs in five main bearings, taking its end-location from the centre one. The bearing diameters rise in steps of $\frac{1}{16}$in, starting at $2\frac{5}{16}$in at the front and being $2\frac{9}{16}$in at the rear.

The bearing lengths are $1\frac{17}{64}$in at the front, $\frac{15}{16}$in for front and rear centre, $1\frac{5}{8}$in for centre main (thrust) and $2\frac{9}{16}$in for rear, these being nett effective figures. The crankpins are 2in in diameter and the overall length of the journal is 1·212in, thin-wall bearings being used throughout. The stamped steel connecting-rods have $7\frac{5}{8}$in centres, and the bolts are fitted with separate locknuts of the Palnut type. Fitting instructions call for 40-45 lb ft torque on the main nut and only sufficient to lock on the Palnut. No split-pins or tab-washers are used.

Clamped in the small ends of the rods, the gudgeon pins are 0·8126in diameter and work direct in the anodized full-skirt aluminium alloy pistons. Two gas rings are employed, these being recessed at the inner top edges to provide a slight twisting action and a moderate scraping effect. Below these there is an orthodox slotted scraper ring and below this, a special channel-section ring of pressed steel, cut nearly right through at a number of points alternately from above and below, giving a zig-zag formation. Ring pressure comes from the circumferential compression of the "zig-zags," the ring ends butting. This ring, which goes by the name of Flex-Fit, appears to be something new. It has the advantage of giving absolutely uniform pressure while being very flexible.

The piston crowns project into the combustion chamber in a shape which gives a slight "squish" turbulence on one side and concentrates the bulk of the charge on the other side in an approximately spherically shaped space.

Camshaft and valve gear

Carried in five thin-wall bearings stepped down in diameter from about $2\frac{3}{16}$in at the front to $1\frac{3}{4}$in at the rear, the stamped steel camshaft is driven by a Link-Belt silent chain of the inverted-tooth type. The 1in diameter tappets have inside them the plungers of the automatic clearance adjustment, with a return spring and a ball valve in a retaining helmet which limits its lift.

Oil supply comes by gravity down the hollow push-rods, which are fed from the rocker shaft through drillings in the rockers communicating, via an annular space and a radial hole, with an axial hole in each ball-pin.

The exhaust valves have a throat diameter of $1\frac{3}{16}$in and inlet valves $1\frac{3}{8}$in, the lift is 0·348in. A slight venturi is machined in the valve seat behind the throat so that the diameter at approach is less than that given for the throat. Dual valve springs are fitted, with conical split cotters, and the rockers are drilled to feed oil to the valve stems, which have no oil retaining washers.

Manifolds and carburetter

A dual inlet manifold is used, in conjunction with a twin-throat carburetter, the centre four cylinders being fed as one unit and the two end pairs as another. The exhaust manifold is a separate casting, bolted to a T-piece, which is attached by a horizontal face to the underside of the inlet manifold. A rotating shutter operated by a bimetallic spring, diverts all the exhaust gases around the T-junctions of the inlet manifold until the engine is partly warmed up.

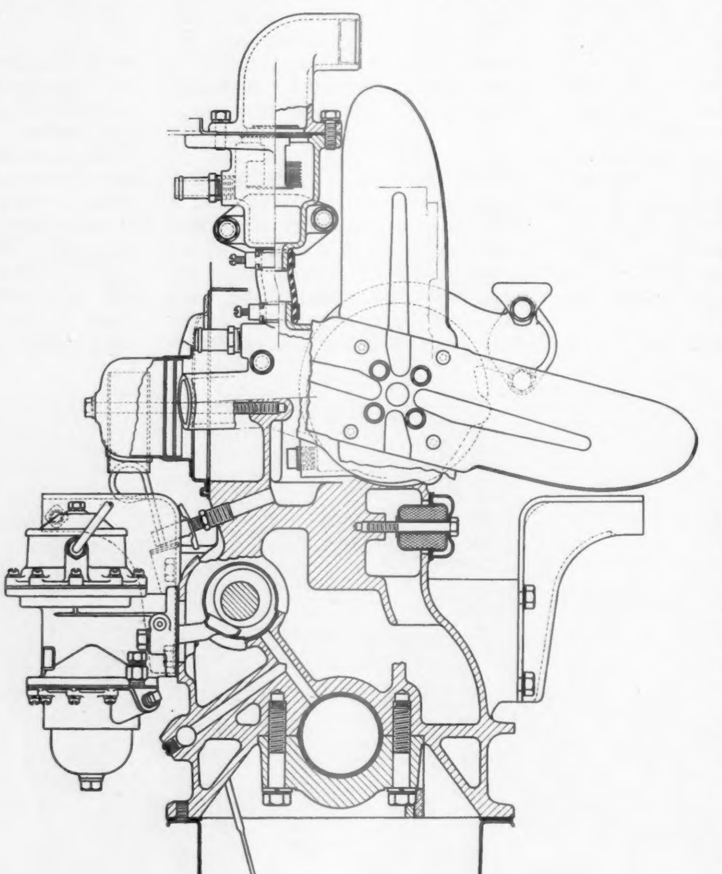
Either Stromberg or Carter carburetters are fitted, both having accelerator pump, automatic chokes with fast idling, and vacuum switches for starter operation by the accelerator pedal. An A.C. heavy duty oil bath air cleaner is connected to the carburetter via an A.C. air silencer.

Lubrication

The gear-wheel oil pump has a piston relief valve in the pump body, and it draws its oil through a floating intake and delivers to an oil gallery placed low down in the block. Diagonal holes from the gallery make a T-junction with diagonal holes joining the main and camshaft bearings. Feed to the valve rockers and tappet adjusters is through an A.C. by-pass filter having a relief valve which opens if the filter gets clogged and ensures supply to the tappets at all times.

Cooling

The centrifugal water pump has an impeller $4\frac{1}{4}$in in diameter and a rubber-mounted carbon ring seal in a brass housing, that has lugs engaging keyways in the ring



Engine cross-section showing main bearing.

to prevent rotation. The fan spindle is integral with the inner race of the double-row ball bearing and is driven at about 0·8 of engine speed by a vee-belt, triangulated over the dynamo pulley.

The fan is mounted lower than in previous models and is 18in. in diameter. It has four unequally-spaced blades about 3in wide with a twist of $1\frac{7}{8}$in instead of the previous $1\frac{5}{8}$in to compensate for the reduced speed at which it is now driven. Temperature control is by thermostat in the outlet connection, with a $\frac{3}{4}$in by-pass pipe.

The cellular radiator has a block 2in thick. When the synchromesh gearbox is fitted the block has a frontal area of 415 square inches. When the Dynaflow converter is employed the radiator block is increased to 485 square inches to allow for the heat rejected to the engine water by the converter oil cooler.

Ignition and starter

The distributor of the coil ignition has both centrifugal and vacuum advance and a single contact arm of exceptionally light construction working on an eight-lobed cam. The spark plugs are 14 mm, normally set to a gap of 25/1,000in.

Following Buick previous practice the starter has solenoid engagement before the main switch is closed. The sliding pinion is actuated through a spring and has a roller clutch to prevent overspeeding of the armature when the engine fires. Operation of the relay is by the accelerator pedal, with cut-out by build-up of generator voltage when the engine fires.

The generator has both voltage and current control and gives a maximum current of about 40 amperes. It is driven considerably above engine speed and is rated for speeds up to 8,000 r.p.m.

Torque converter

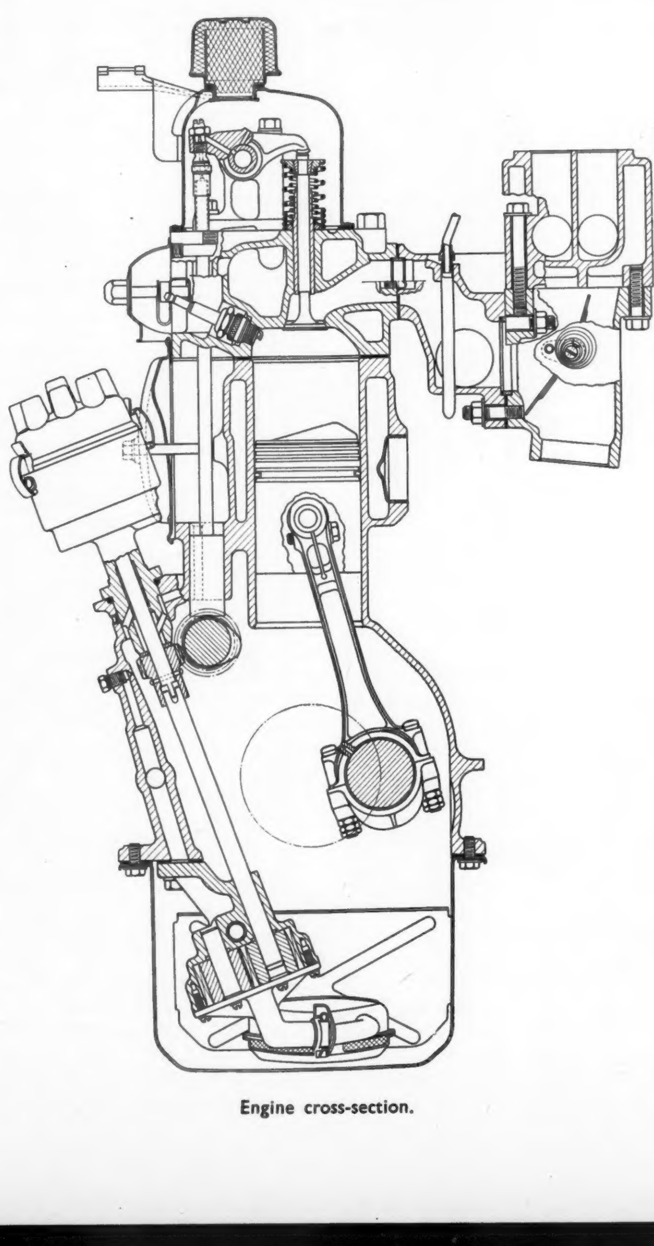
Bolted to the crankshaft flange is a light pressing, to which the starter ring is welded. The complete converter assembly is attached to this pressing by a ring of bolts and can be removed as a unit without exposing the internal parts. The light pressing also gives a slight universal joint action to allow for small misalignment of engine and transmission units.

The front enclosure of the converter is a pressing, housing at its centre a spigot ball bearing supporting the input shaft. The rear portion is formed by the aluminium-alloy pump casting which has riveted to it a steel sleeve called the "pump hub" which extends through an oil seal into the front of the transmission case. There it picks up, by two dogs, the pinion of the front oil pump supplying oil for controlling the planetary gears in the transmission case. Mounted on a roller free-wheel on a forward extension of the pump hub is the secondary pump of the converter; this can overrun the pump.

The turbine is riveted to a stamping splined to the input shaft and between it and the secondary pump there are two stators, called secondary and primary. These are carried by roller free-wheels splined to a sleeve, called the reaction shaft, which passes through the pump hub and is secured to the "reaction flange" bolted to the transmission casing. The free wheels allow the stators to overrun in the direction of engine rotation but prevent backward movement.

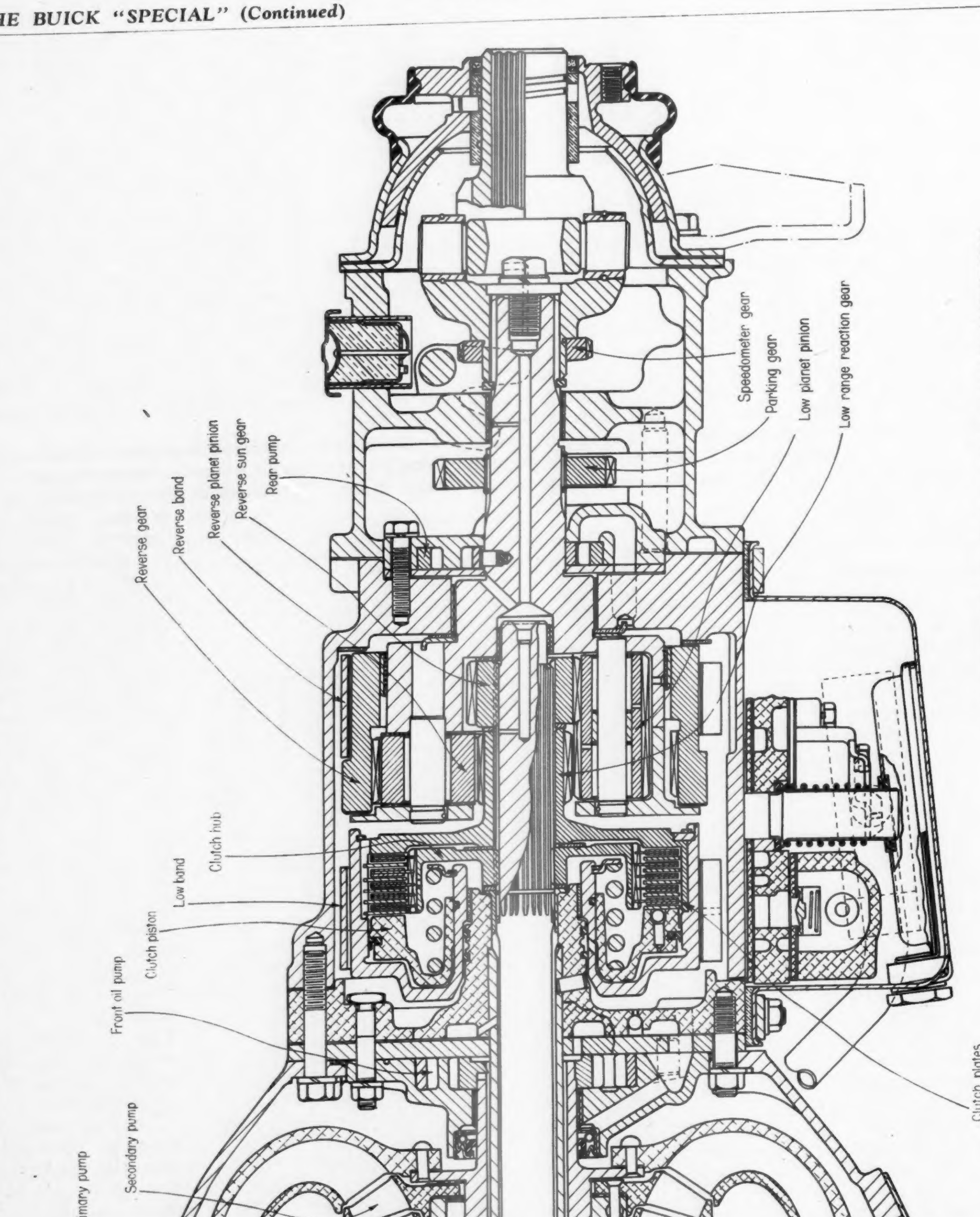
When starting from rest and at low speeds when the turbine is practically at a standstill, the difference of speed between it and the converter pump is considerable, and the resulting vortex flow very rapid. The vanes of the turbine at their inner ends are curved backwards and the reaction of the oil issuing at high velocity causes a considerable torque on the turbine, reaching a maximum of 2·25 times engine torque. But since the issuing oil opposes the motion of the pump a correspondingly high torque would have to be applied to the pump, in the absence of the stator vanes. The mechanism would in fact be merely a special type of fluid flywheel.

The purpose of the stator vanes is to turn the backwards flow of the oil and direct it into the pump in a more or less forward direction, thus assisting the pump and reducing the torque to be supplied by the engine. In the final result the turbine torque is equal to engine torque plus the reaction torque taken by the stator vanes. The working may be a little difficult to visualize since the stator vanes come

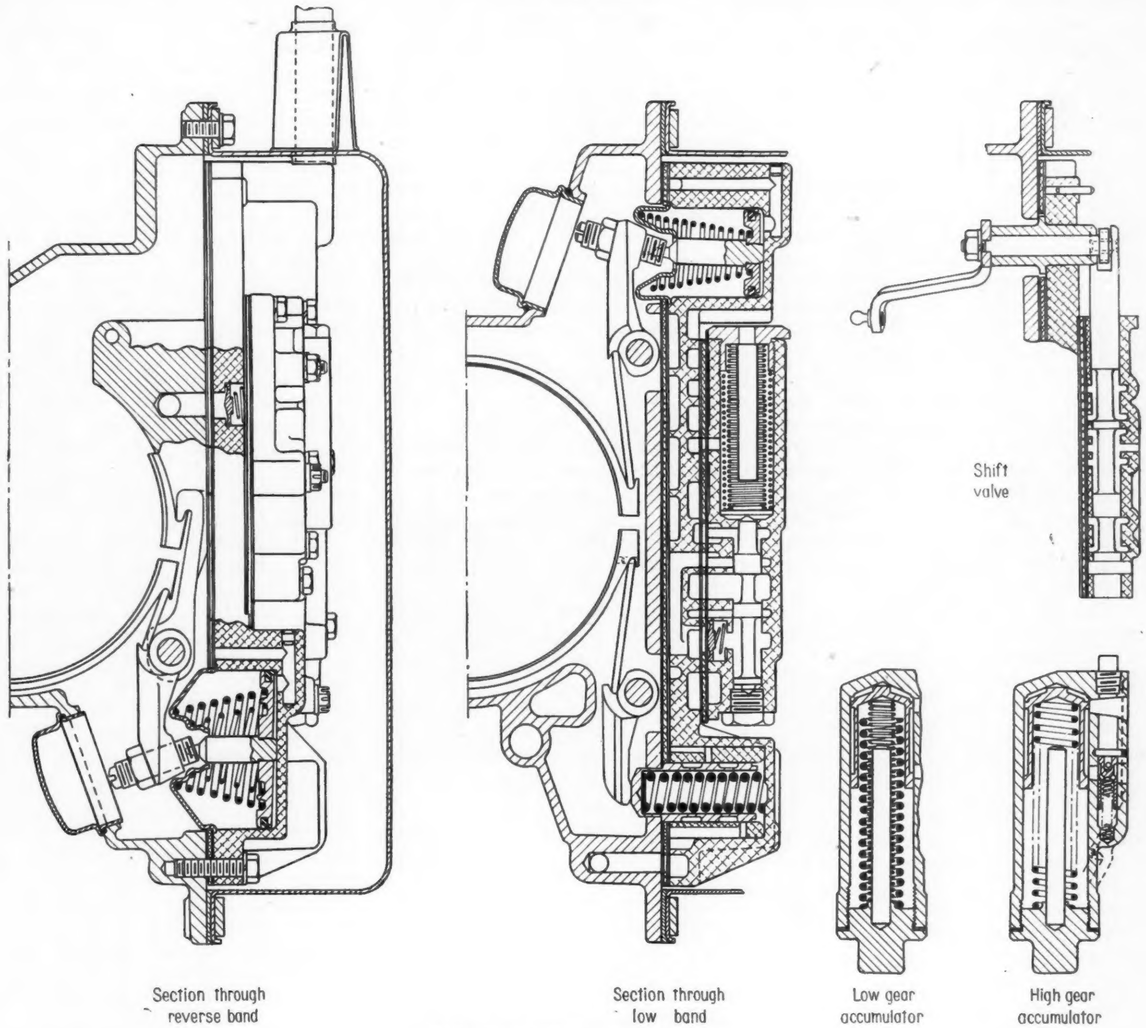


Engine cross-section.

THE BUICK "SPECIAL" (Continued)



LONGITUDINAL SECTION OF DYNAFLOW TRANSMISSION.



Part cross-sections of gearbox.

after the turbine and therefore have no *direct* influence on the turbine torque.

The stator element is made in two sections, allowing one to overrun on its free-wheel when the resultant flow is at such an angle that the oil strikes the backs of the vanes. As the turbine speed rises and the difference between pump and turbine speeds falls, the vortex flow is reduced until a point is reached when the second stator also becomes superfluous, overruns on its free-wheel and the mechanism becomes a fluid flywheel with no torque multiplication.

To assist in matching the blade angles to the resultant flow into the pump under varying conditions, the pump is also subdivided. The secondary pump has vanes of a steeper angle than the primary. When the flow approximates to the primary blade angle, the oil striking the backs of the secondary pump vanes causes them to overrun on the free-wheel without obstructing the flow. This action takes place at low turbine speeds and high oil flows. At higher speeds and reduced flow, the secondary and primary pumps run as a unit with the vanes forming a more or less continuous curve.

It should be pointed out that no compensation by modifying blade angles is necessary at the change over from pump outlet to turbine inlet. When there is great difference between pump and turbine speeds the oil flow is correspondingly high and keeps the resultant angle constant.

The efficiency of the converter varies from zero at 2·25 to 1 multiplication to about 97 per cent at full engine speed and no multiplication, the deficit representing the "fluid flywheel" slip. Efficiency rises sharply from the stalled figure, reaching about 70 per cent at 1·5 to 1 multiplication and about 90 per cent at the point where multiplication is low and the change to "fluid flywheel" conditions is imminent.

On full throttle every engine speed corresponds to a definite conversion ratio, the drop from unity beginning at about 2,500 r.p.m. At 2,000 engine revs, and full throttle, the torque multiplication is about 1·5 to 1 and the propeller-shaft speed about 1,100 r.p.m. This would correspond to climbing a gradient of about 1 in 7.

It will be clear that, in a car with such a good power-weight ratio as the new Buick, the rather low efficiency of the converter, as compared with second speed of a three-speed gearbox, is of no practical importance in ordinary driving. In acceleration, it is more or less compensated for by the continuous change in ratio, unbroken by the pause involved in changing up on an orthodox gearbox. Fuel consumption is not much greater than with the standard three-speed gearbox, partly because it has been possible to raise the compression

THE BUICK "SPECIAL" ***(Continued)***

ratio where the Dynaflow converter is used, since the engine runs faster when accelerating from a fairly low speed and is hence not so prone to pinking.

Transmission

To provide a low gear for exceptionally arduous conditions and to give better engine braking when descending steep gradients, as well as to give reverse motion, a planetary two-speed and reverse transmission is provided. Engagement of gears is by a manually selected hydraulic system. The driving sun wheel is splined to the turbine shaft and meshes with three planet pinions each meshing with an idler, these in turn meshing with a sun wheel, which, when held stationary gives a low forward gear of 1·82 to 1. The first three pinions also engage an internal gear, which, when held, gives reverse at the same ratio as low gear.

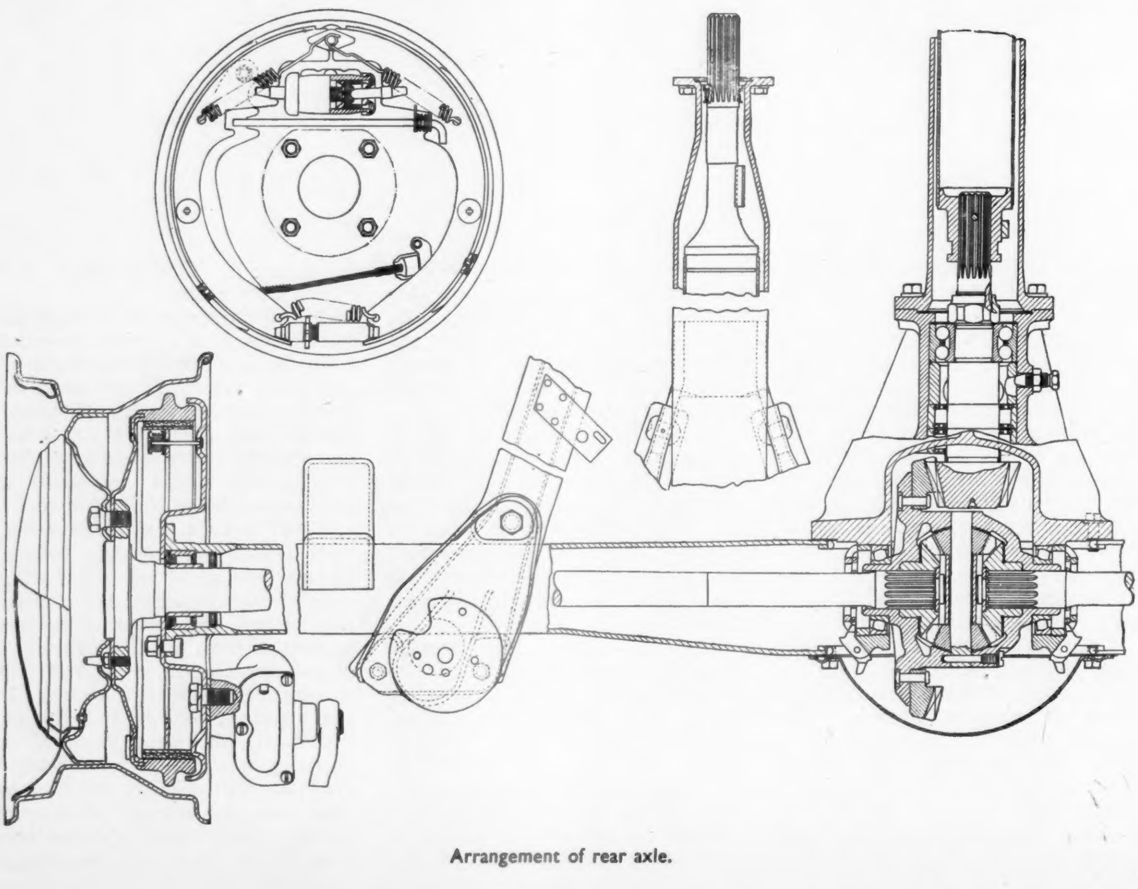
The planet carrier is built up of two steel stampings, the rear one of which is extended to form the output shaft and carries the single universal joint in the torque-tube ball. Direct drive is got by a multiple-disc clutch locking the sun wheel to the turbine shaft. This is applied by direct hydraulic pressure, the outer drum of the clutch containing an annular piston bearing direct on the clutch discs. Oil supply is by a drilling in the projecting neck of the reaction flange, on which the hub of the clutch drum is supported by a bush. Sealing is effected by two piston-rings, one on each side of the feed hole.

The two band brakes for low gear and reverse are applied by struts and levers operated by servo cylinders having neoprene seals to their pistons. The planet pinions run on needle roller bearings, but every other part of the transmission, except the turbine spigot bearing, runs on plain bushes, all provided with forced lubrication. The output shaft is carried in two bearings, between which is the rear oil supply pump and a toothed wheel with which the parking lock pawl engages. Behind the rear bearing is the speedometer drive and the plain bush Spicer-type universal joint, which also has positive lubrication.

The hydraulic system merits careful study. To those whose first impression is one of extreme, and perhaps unnecessary complication, attention should be drawn to the many functions it sets out to perform. To begin with a large volume of oil is available at high pressure for operating the low gear and reverse bands and at lower pressure for direct drive. This oil supply begins as soon as the engine starts, but a second, smaller pump is also provided, driven from the rear of the transmission, which will supply sufficient oil to engage the clutches for starting the engine by towing the car.

The large pump, which would waste a lot of power if continually in use, is automatically put out of action when the working pressure is reached, leaving the small rear pump to maintain the pressure. The large pump, is of course, instantly available when required.

Engagement of clutch and low gear is arranged to take place rapidly up to the point of taking up clearances, after this the engagement pressure is increased more gradually to give a smooth take-up. Special mechanism is provided to ensure that the engine does not "race" when changing from low to direct drive, the low gear band not being



Arrangement of rear axle.

fully released until the direct drive clutch has begun to take hold.

The system provides for maintenance of a pressure of some 35 lb per square inch in the converter to prevent cavitation and circulates this oil through an oil cooler to prevent excessive rise of temperature. Oil at a reduced pressure is also fed to every bearing in the transmission, including the universal joint.

As an example of careful attention to detail, there is the fitting of a ball valve in the rotating clutch operating cylinder, which prevents oil pressure being built up by centrifugal force when the clutch is supposed to be "free," coupled with the fitting of a second ball valve at the clutch supply point to admit air and prevent the escaping oil from the cylinder sucking the associated control passages dry and causing inconsistent operation.

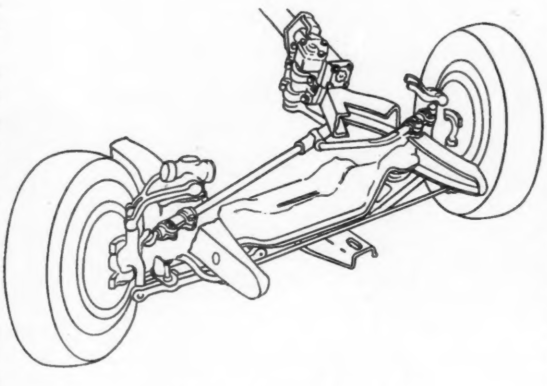
Oil pumps

These are both of the internal gear type. The front pump is driven by tongues on a rearward extension of the converter pump hub, the pinion of the smaller rear pump being driven by a peg on the output shaft. Both draw through a fine gauze screen from a pressed steel sump, and deliver, through check valves, to the space between the first and second lands (starting on the left) of the pressure regulator valve.

Pressure regulator valve

This is held to the left by two low-rated compression springs. On starting the engine the front pump rotates immediately, being coupled direct to the engine flywheel. It lifts its check valve which is actually in the regulator valve body and builds up a pressure behind the second land, the first land being merely a dashpot with a bleed hole. The valve moves to the right, uncovering first the port connected to the converter feed.

Oil passes through a bleed hole into the converter along the annular space between the converter pump hub and the reaction sleeve. It returns along the space between the reaction sleeve and the turbine shaft until it is stopped by a piston ring in the shaft and passes by a side hole to the oil cooler. From this it escapes through a relief valve loaded to about 30 lb per square inch. A bleed hole in the relief valve passes some of the oil at the reduced pressure of 15 lb per square inch into the various lubrication passages by which every bearing in the transmission is oiled. It will, therefore, be seen that before anything else happens pressure is supplied to the converter to prevent incipient cavitation.



Layout of track rods.

What is equally important is that the converter feed port closes when the engine is switched off and comes to rest, locking the oil up in the converter and preventing seepage and entry of air. The pressure in the regulator valve body continuing to rise as the engine speeds up, carries the valve still further to the right until its third land uncovers a port by-passing the front pump to the sump via the chamber between lands 2 and 3. This position corresponds to a pressure of about 90 lb per square inch, which is the working pressure for the direct drive clutch.

Assuming that the transmission has been engaged and the car is under way on direct gear, the rear pump will also be delivering, and, if there are no demands for clutch operation, pressure will be further built up by the rear pump, opening the front pump bypass fully so that it is now idling. Pressure is now maintained at a slightly higher value, relief being by the second land on the valve running out into the sump return chamber.

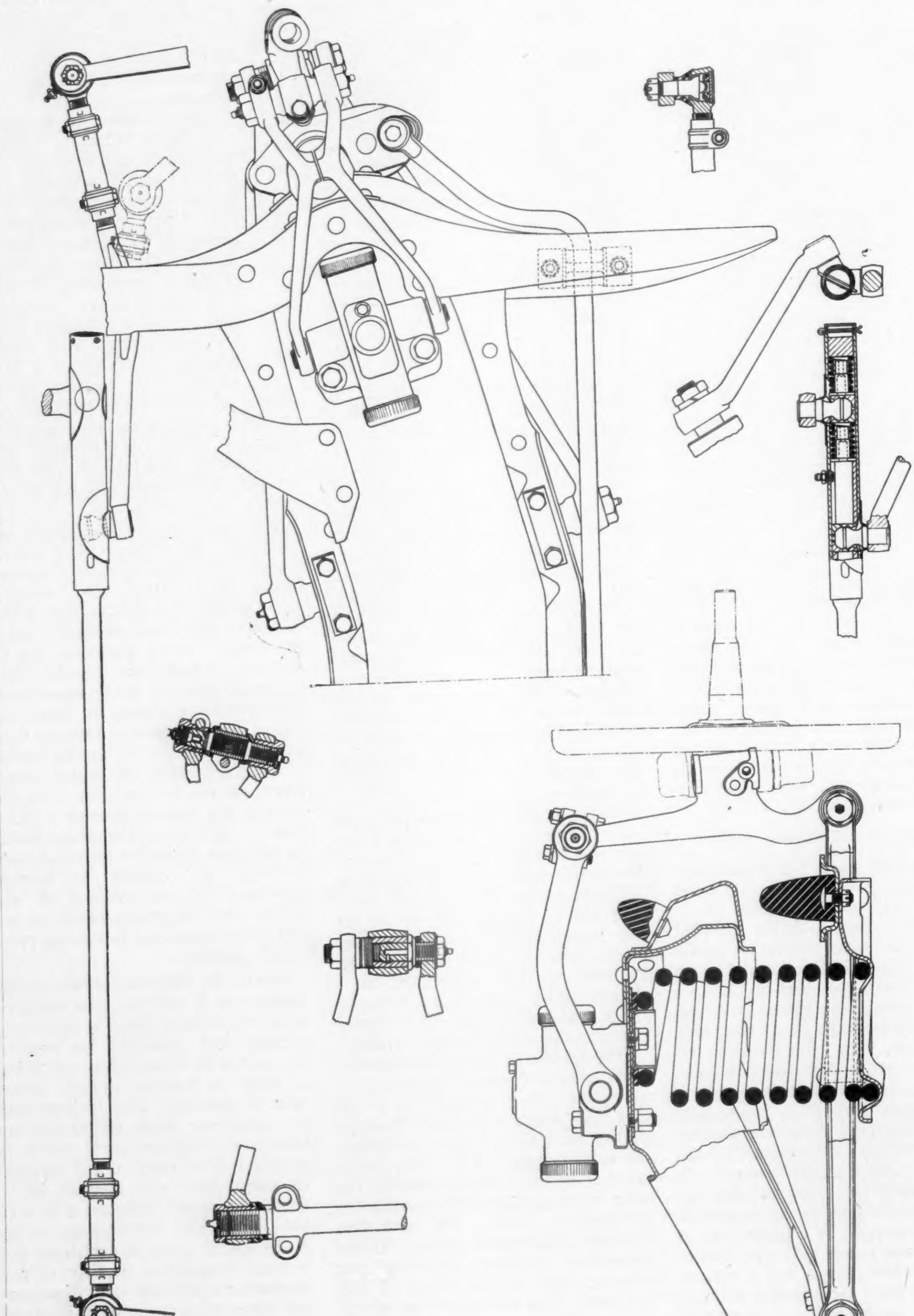
The spindle of the regulating valve is reduced in diameter to an amount giving an area of one-half the piston area and works in a guide forming part of the sealed chamber containing the valve springs. When low gear is selected by the manual control, the shift valve connects this chamber to the pressure circuit. The valve then becomes a differential piston, with the controlled pressure opposing the opening force to the extent of half the total. The net area being halved, the working pressure is consequently doubled, giving the 180 lb per square inch necessary for the low gear operation. The same pressure is also required for reverse and the shift valve connects the pressure circuit to the annular space formed by the last land and the spindle guide, giving the same net area and the same pressure increase.

Since reverse is engaged with the car at rest, no further refinements are necessary in order to secure cushioned engagement, but for both low and direct drive engagement additional mechanism is introduced. This takes the form of what is described in the service manual as an "accumulator," a term employed here for consistency's sake while pointing out that it is a misnomer. It implies some storage of fluid preparatory to an operation, whereas the exact opposite is the case. Each "accumulator" has a long piston with lands at each end normally held against the top of the cylinder by a spring.

When a gear is to be engaged, oil from the shift control valve passes round the reduced neck of a spring-loaded valve, called the "dump" valve, and into the cylinder just above the lower piston land passing up along the "waist" of the piston and out to the selected application cylinder. The oil pump delivery being very large, the working clearance is taken up almost instantaneously. Pressure then tends to rise but the spring-loaded piston gives way to this and is forced down, closing the top port. All the oil now has to pass through a bleed hole in the neck of the dump valve, so that flow is greatly restricted, and build-up of pressure is further graduated by the yielding of the spring until the piston comes to the end of its travel and full pump pressure is applied.

One of the valuable features of this mechanism is that the time taken to secure full engagement is practically constant and unaffected by wear of the clutch or brake while yet being as short as possible without fierceness of take-up. This is important in connection with the change up from low to direct drive which is intended to be made at full throttle. In conjunction with another mechanism it enables direct drive to take hold under load without any racing of the engine at the change-point and without momentary locking of the transmission although both direct and low gears are, for an instant, in partial engagement at the same time. Release of the gear is rapid, the dump valve lifting and allowing the oil to pass direct to the sump while the accumulator piston is returning to its un-

THE BUICK "SPECIAL" (*Continued*)



ARRANGEMENT OF INDEPENDENT FRONT SUSPENSION.

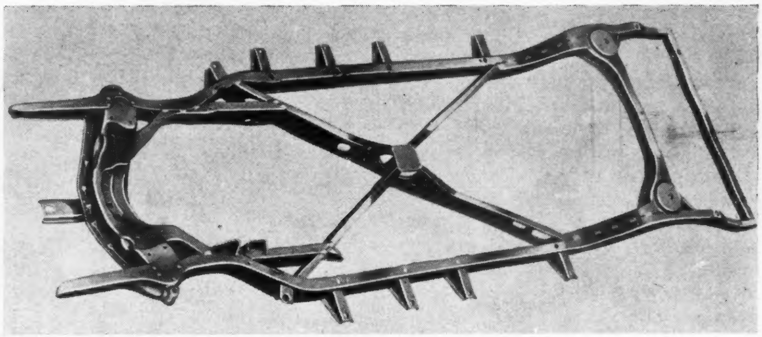
loaded or inoperative position.

The low gear band is applied by a piston operating a bell-crank and strut. It is anchored by a similar bell-crank the long end of which bears on a piston normally forced against a shoulder by a strong spring. The piston has also three lands and controls several oil passages. On starting in low gear the anchor piston is up against its shoulder and oil from the low gear apply port on the shift control valve passes round the upper waist to a port on the right-hand side connecting with the low gear application piston via the low accumulator. As the band takes up the drive, the reaction forces the anchor piston right down, cutting off the low gear apply port but opening a second port on the right-hand side connected to a second port on the shift control called the low relay port. By this the low gear piston continues to get pressure from the oil pumps.

On shifting to high gear a land on the control valve vents the low gear apply, and the low gear boost chamber to the sump. Low gear is still held in by the relay supply and the top land on the anchor piston prevents the low gear piston venting to sump. Working pressure drops to 90lb per square inch on release of the low gear boost pressure. Simultaneously the second land on the shift control valve opens the supply to the high gear accumulator and clutch piston, including also a passage, via the anchor piston, which by-passes the accumulator.

As the high gear clutch begins to take up, the reaction on the low gear band drops allowing the anchor piston to rise, venting the low gear piston to sump via the low gear apply port on the shift valve which is already open. The effect is that while both gears are momentarily engaged at once, the low gear band has had its grip halved by the reduction of working pressure. As the clutch takes hold, the resulting release of low gear band reaction frees the low gear band altogether. Racing of the engine is prevented and yet the low gear band is at no time "holding back" the high gear.

The shift control valve has a neutral position, in which pressure is held at 90lb per sq in to keep the converter charged, but all operating pistons are vented to sump. In yet another position marked "P," the same conditions obtain but a pawl is engaged with a gear on the output shaft to lock the transmission. In all but these two positions the starter circuit is broken by a safety switch, so that the engine cannot be started unless the transmission is in neutral.



The Buick "Special" frame.

Back axle

Having a ratio of 3·9 to 1, the hypoid gears are mounted in a cast-iron housing. The pinion is carried by two Hyatt solid roller bearings having straight-through races with sprung-in locating rings, so that one-piece pressed-steel cages can be used. End thrust and outer support is provided by a double-row New Departure ball bearing. The differential-box is carried on two angular contact bearings having barrel-shaped rollers.

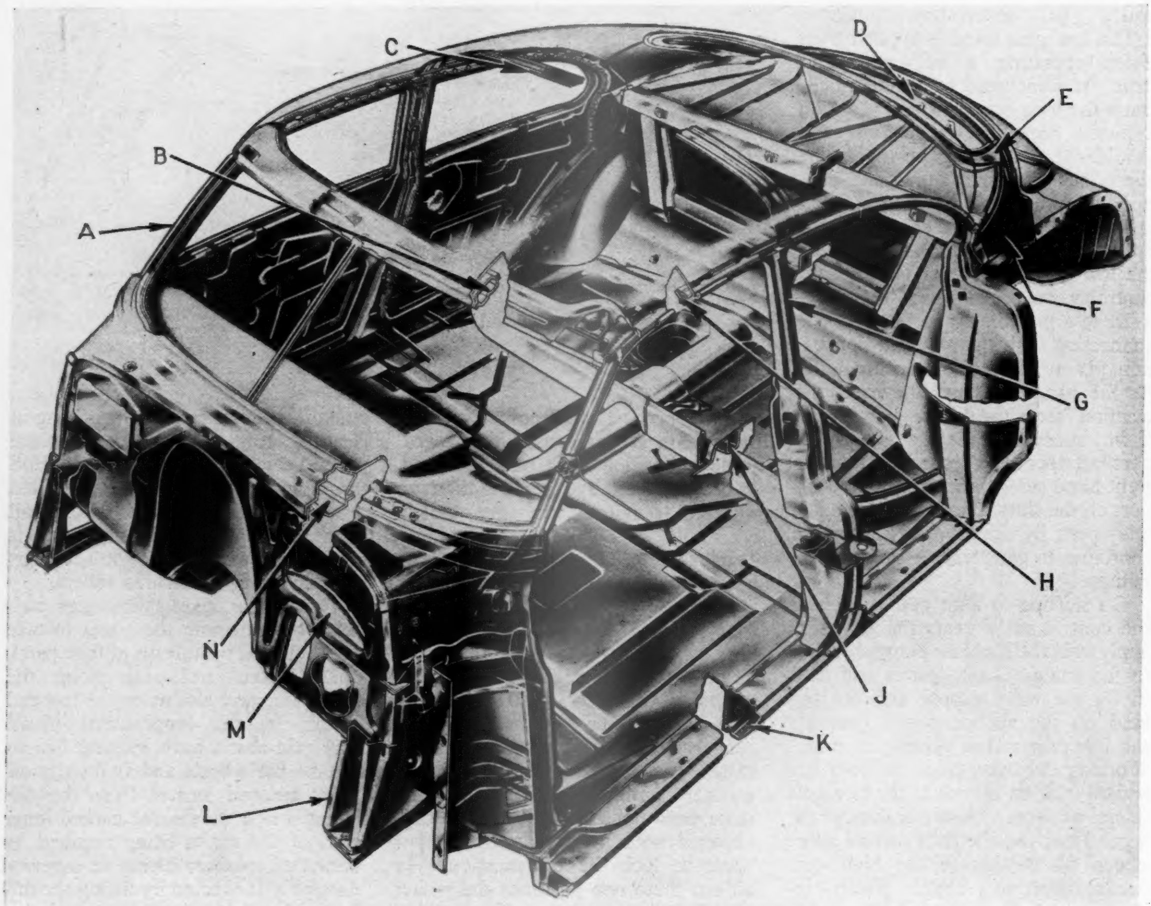
The banjo has its openings stiffened by inturning the edges to take the studs. It is built up of four parts, welded front and rear along the centre line and also at centre top and bottom in the longitudinal plane. The axle shafts have integral flanges to take the wheels and brake drums. They are end located from the differential side wheels, the necked inner ends of the shafts being retained by horseshoe washers fitting in recesses. Assembly is effected by fitting the differential cross-pin, pinions and distance-piece as a last operation on the assembled axle.

The torque tube has integral flanges and is stayed by two trough-section diagonals. These are riveted at their front ends to lugs on a pressing welded to the torque-tube and clipped to the axle arms at the rear by bolts and pressings. These extend rearwardly to form seats for the rear suspension coil springs. They have the end coils reduced and secured to the pressings and in pockets in the frame by washers and bolts. Lateral location is by a tubular Panhard rod with Harris-type rubber bushings. The rod passes at an angle from lugs attached to the offside torque-stay clip to a bracket depending from the rear spring cross-member on the near side.

The Buick "Special" 43 four-door saloon.

Frame

The channel-section side members are parallel for most of their length, but are sharply inswept at the front to provide the wheel lock. They curve outwards again to embrace the ends of the front suspension cross-member and for the attachment of the front bumpers. A simple and effective "X"-bracing is built up of



General view of body structure.

A. Narrow screen pillar. B. Box-section windscreen frame. C. Roof bow 4in wide. D. Reinforcement strengthening rear window section. E. Rear window frame. F. Double walled construction. G. Narrow centre pillar. H. Box-section roof rails. J. "Hat"-type cross bars forming box sections. K. Box-section rocker panels. L. Brace between dash and frame. M. Access hole to central fuse block. N. Integral de-froster duct.

four channel section arms welded to top and bottom plates with additional web reinforcement. Since the junction is under the front seats it has been possible to make the overall depth nearly 9in at this point. This arrangement, together with the fact that the four arms have a common intersection point makes for a stiff bracing subject to simple stresses.

The side members are $5\frac{9}{16}$in deep and have flanges $2\frac{1}{4}$in maximum width, the thickness being $\frac{7}{64}$in. The inswept portion is reinforced by submembers welded to the front X arms. and to the side member flanges. They then curve inwards and are welded as diagonal bracing to the front suspension cross-member, which is of the customary inverted trough section.

At the rear the frame is reinforced by channels following the curve and tack-welded to the frame flanges to form a box section that picks up the ends of the rear X bracing arms. The reinforcement extends rearwardly to an inverted trough-section cross member, the ends of which are widened to embrace the rear coil springs which take a bearing in circular recesses. A light channel member at the extreme rear completes the frame. There are five body outriggers on each side and a number of additional attachment points on the side-member flanges. Frames intended for open bodywork have thicker subrails, rear reinforcements and X-member cover plates, with additional channel reinforcement of the webs of the arms round the junction point.

Closed bodies are attached by bolts with rubbber grummets on each side of the flange, giving two-way cushioning, but open bodies are secured with bolts having rubber on one side only.

Front suspension

Of orthodox coil-spring and wishbone construction the front suspension follows what may now be considered to be standard practice throughout the American industry. The lower wishbones are $19\frac{3}{4}$in centres and the side stampings are riveted to the spring pans. Case-hardened steel threaded bushes are used for all the joints, the threaded outer diameter cutting its own bearing in the plain bored eyes, the assembly shop torque figure being about 100lb feet. Adjustment for camber is provided by an eccentric centre portion of the top joint pin, which also gives adjustment for caster by turning through complete revolutions as necessary.

Normal settings are about 1 deg positive caster, zero camber and $4\frac{1}{4}$ deg swivel pin inclination.

The steering arms are $8\frac{1}{2}$in long and this, together with the long lower wishbones reduces to a negligible degree the slight errors of geometry of the simple steering linkage. In this a long rod runs from the arm of the

steering box to the opposite swivel and has, at an intermediate point, a socket from which a short rod runs back to the near swivel. The steering column has a very flat rake and the rise and fall of the drag-link ball on lock is slight. The steering gear is of Saginaw make, and has a nut carried on two sets of circulating balls and operates the steering spindle by a rack. The teeth on the shaft are cut on a bevel so that axial adjustment of the shaft will take up backlash.

The usual stabilizer bar is mounted in rubber bushes and has rubber washer linkage to the wishbone arm. No rubber is used anywhere on the load-supporting wishbone linkage. This point is significant in view of the extended experience of the American industry in the use of wishbone suspension, coupled with the development work they have done in the use of rubber for rear axle leaf spring linkage.

Body

The Fisher bodies evidence many changes, notably in increased width assisted by the moving of the back seats clear of the wheel arches. It is interesting to note that a slight propeller shaft tunnel is accepted as a means of reducing overall body height. The curved windscreen is centrally divided and is very much wider than before, while the pillars are narrower and have quite small radii at the corners. At an admitted sacrifice of body stiffness, the driving visibility has been immensely improved. Some stiffness has been recovered by the use of box section roof rails and front rail. The rocker panels are also of box section while the floor pressing has five cross-bars and is stiffened by footwell indentations besides the usual ribbing. The overall length of the car has been reduced by 4 inches by combining the front grille with the bumper, the chromium-plated guard pressings on which are sufficiently numerous to give a grille effect.

Following standard American practice the windscreen wipers are suction operated, a single motor working the two wipers by cables passing over pulleys. The A.C. fuel pump has the usual vacuum booster pump incorporated to keep up the wiper speed when the manifold depression is inadequate. Trafficators, as understood in this country, are not used in America, their place being taken by optional flashing light signals at front and rear. Cancellation by steering wheel movement is provided.

Dimensions: —Wheelbase 121½in. Front track 59·1in. Rear track 62·2in. Overall length 204in. Overall width 79·4in. Overall height 63·9in. Turning circle 39ft 6in. Kerb weight 3,900lb. Tyres 7·60–15 (wide-base rims). Ground clearance minimum (frame side rail) 6·82in.

WILLYS ENGINE DEVELOPMENT

A Four-cylinder F-head Type Introduced

IN announcing the 1950 range of Willys vehicles, Willys-Overland Motors Inc. are offering a new four-cylinder engine for certain models, according to *Automotive Industries*. The new engine, which is of the F-head or inlet over exhaust type, is basically the same as the original side valve four-cylinder engine fitted in the well-known Jeep, and it is notable as a development of design resulting in the minimum of investment in new tooling and machinery. The cylinder block, connecting rods, pistons, crankshaft and many other parts remain interchangeable with the side valve engine and the only change below the head is the use of a new camshaft embodying new cams for the inlet valves. Although the exhaust valves remain the same in basic design and in timing, "free" type exhaust valve rotators and hard-facing of the exhaust valves have been incorporated.

The new engine is offered with a choice of three compression ratios, namely, 7.4:1 as standard, 6.9:1 for export and 7.8:1 for high altitude conditions. The compression ratio of the side valve engine remains at 6.48:1. The new engine delivers 72 b.h.p. at 4,000 r.p.m. compared with 63 b.h.p. at 4,000 r.p.m. for the side valve design, the relative specific fuel consumptions being 0.49 lb per b.h.p. hour against 0.575 lb per b.h.p. hour.

A feature of the F-head design is that the inlet manifold passages are cast inside the head with the carburettor mounted directly on to the head. The inlet passages are therefore completely surrounded by water jacketing and are effectively heated throughout the range of operation. With the elimination of hot spot heating a definite gain in volumetric efficiency is claimed. The combustion chamber is of asymmetric form with a suitable quench area and the sparking plug is inclined and is located almost immediately over the exhaust valve. The inlet valve is also slightly inclined to the vertical axis and is 2in diameter. This has provided such a large area at the inlet port that it is possible to reduce the valve lift. (1894)

MACHINING ZINC ALLOYS

ZINC alloy die-castings are produced to close limits of accuracy, but nevertheless there are many castings on which some form of machining is necessary. Although the amount of metal to be removed is usually small and the metal is free-machining and comparatively soft, it is still important that the correct practice be employed. The correct procedure and methods are described in a small information book, "Machining of Zinc Alloy Die-Castings," recently published by the Zinc Alloy Die Casters Association, Lincoln House, Turl Street, Oxford. It deals succinctly, yet thoroughly, with every type of machining there may be need to carry out.

SPRING MATERIALS

A 28-PAGE booklet recently issued by Henry Wiggin & Co., Ltd., Wiggin Street, Birmingham, 16, deals with Nickel Alloy Spring Materials. Apart from several pages of design formulæ of general application, there are sections covering the design of springs in high nickel alloys for various conditions of service, such as high temperature, sub-zero temperature, damp or corrosive conditions, etc. An interesting section deals with properties of materials including the Nimonic series of alloys for springs operating in high temperatures, and there are also notes on manufacture and heat-treatment, besides much useful tabulated data. (1893)

ALUMINIUM ALLOYS

THE ALUMINIUM DEVELOPMENT ASSOCIATION, 33, Grosvenor Street, London, W.1, have recently issued "The Application of Aluminium and its Alloys to Passenger Road Service Vehicles—An Introductory Survey." Although primarily directed to those responsible for the running and maintenance of fleets of vehicles, there are many notes and illustrations of interest to the designer. There are four main sections of the book describing and illustrating various types of vehicle, body and engine details, and a description of the commercial forms of aluminium and its alloys with reference to mechanical properties. (1888)

CYLINDER HEAD PRODUCTION

A Review of Recent Developments Employed by Vauxhall Motors, Ltd.

DEVELOPMENT projects costing some ten million pounds are to be carried out by Vauxhall Motors, Ltd., in the course of the next three or four years. They include a new factory building, now nearing completion, with a total floor area of 19½ acres, reorganization of the existing buildings and a great amount of retooling for the production lines. This is all to be carried out without any interruption of production. The new building, which is an essential feature of the reorganization will not be ready until later this year, but it can be said that it will house production lines that are a great advance on those hitherto employed. Incidentally, it is of interest to note that without any major retooling, in fact with approximately the same production facilities and the same labour force, the Company produced 84,000 units in 1949 against an output of some 60,000 units before the war.

Although some months must elapse before the new factory is available for production, some conception of the techniques to be employed may be gathered from a consideration of the lay-out of the machining line for the improved cylinder head now standardized for Bedford truck engines. This line is not yet fully retooled, but even now it incorporates some very important developments. It does however, illustrate the principles that are to be employed, and is in some degree a prototype.

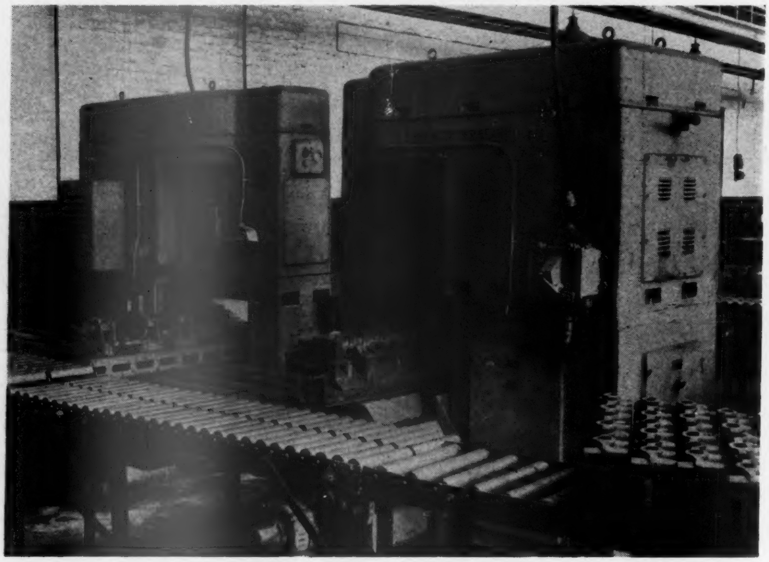


Fig. 1. Archdale special vertical millers for machining rocker and joint faces on cylinder heads for Bedford engines.

Naturally, the first consideration in planning this line was product accuracy, but other important factors had also to be considered. Briefly, these were maximum productivity per unit of floor area and per man hour employed on the line. It is the manner in which maximum productivity per man hour has been obtained that makes this line of absorbing interest to all production engineers, and more particularly to those engaged in the automobile industry.

Obviously the basic factor governing productivity is the actual total machining time necessary to produce the component. This is, of course, governed by the feeds and speeds that can be employed to produce the required standards of accuracy and finish with an economical tool life. Therefore careful planning of the process sequence and the employment of optimum machining conditions are fundamental factors governing the total productivity of the machine line.

Merely to install the best machines and run them at the highest practicable speeds and feeds will not, however, of itself give maximum productivity per man hour employed. Many other factors must also be considered before this goal can be reached. In its simplest terms the problem becomes one of so arranging the work that the machines in the line can be tended by the fewest possible number of operators. This may be effected either by combining operations on a single machine or by having one operator tend several machines. Both methods are used on this cylinder head line.

Automatically controlled machine cycles are, of course, the first necessity when one operator has to serve several machines, but with such cycles loading and unloading times and operational fatigue become important problems. With a relatively heavy component such as the cylinder head for a

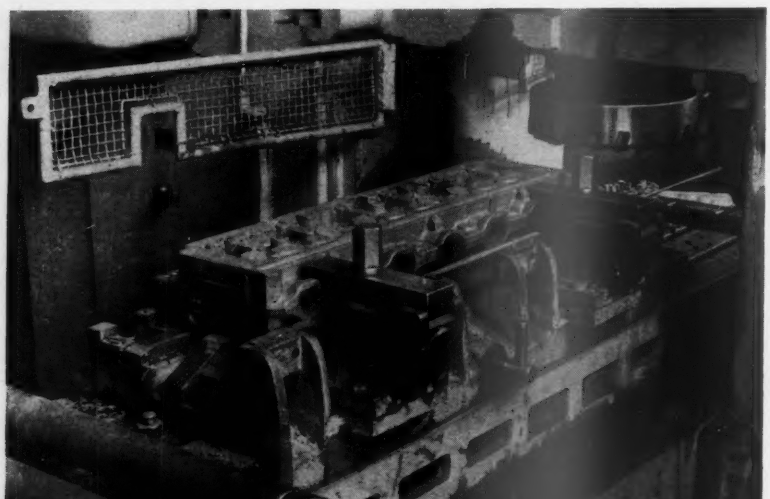


Fig. 2. The set-up for machining the rocker face.

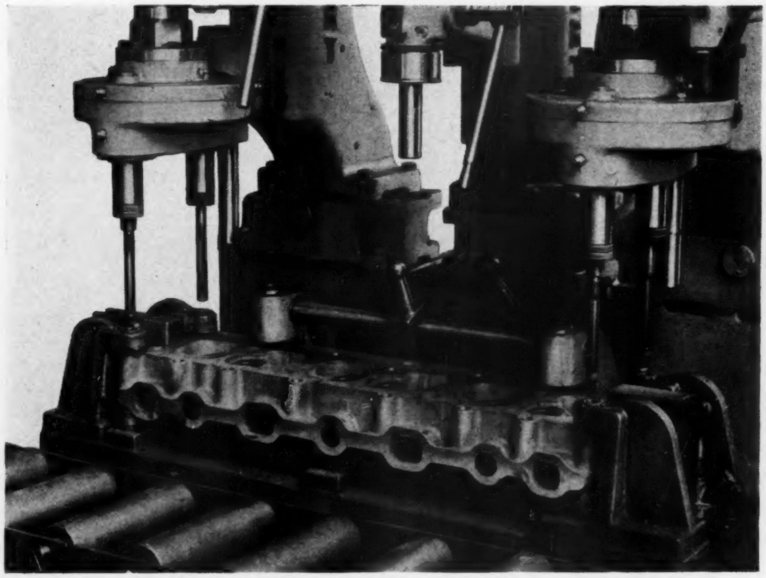


Fig. 3. Reaming locating holes on a Herbert four-speed sensitive drilling machine.

Bedford engine, the main factor in causing operational fatigue is the effort entailed in putting the casting in, and taking it out of, the fixture. Therefore, handling fatigue must be reduced to a minimum. This is effected by arranging the machines and fixtures in such a manner that the operator can slide the casting into the machining position.

Transfer from one machine to another is by means of a roller track and to allow easy loading and unloading, it is necessary to have a standard loading height for all the machines in the line. The machines as received have different loading heights, but the foundation for each machine is so positioned in relation to the level of the shop floor that there is a constant loading height for all machines in the line.

Handling has been further reduced by the use wherever possible of open-ended fixtures. These fixtures are in line with the roller conveyor, so that the casting is fed straight into the fixture at one end and at the completion of the operation it is pushed out at the other end on to the next length of roller conveyor. Ideally, all the machines would be arranged in this manner, and the component would travel in a straight line from the first operation to the last. It has not been possible to lay-out this line for uninterrupted straight line flow of this type, but the principle has been adopted as far as is practicable. Where there is departure from this principle, the machine and its fixture are so arranged that the operator can easily slide the cylinder head from the track into the machining position.

Although in dealing with this type of component, provision must be made to reduce operation fatigue before the number of operators can be reduced, other problems must also be considered. In general, the cycle time for the various machining operations in the complete sequence varies from two to four minutes. Therefore, if an operator is to work more than one machine, the unloading/loading cycle time must be short. That is, the machined component must be released and removed from the fixture, and a fresh one introduced, accurately located and securely clamped in a matter of seconds.

To allow for easy insertion and removal of the component, the most open possible type of fixture has been designed for every operation. In general, the principle employed for locating the work is that the casting is pushed in to the fixture up against a stop that gives approximate location and is then accurately positioned by plungers that register in reamed holes. Apart from the special case of two transfer machines, the locating plungers on a fixture are actuated by a single lever.

Clamping methods vary. Air clamping is used on several machines and where manual clamping is employed, it is generally of such a character that simple manipulation of one, or at most two, levers effectively secures the work. On only one or two machines does an operator use a spanner to tighten the clamps. An interesting point is that on many machines the locating and clamping systems are so interlocked with the electrical controls that the machine cannot be started until the component is accurately located and securely clamped.

The developments described above may be classed as logical developments of conventional practices and techniques. In themselves they would considerably reduce the labour force necessary for a given output. However, in addition to these developments, the line shows a radical departure from standard practice in that it includes two specially designed transfer machines.

It is well known that increasing use is being made of transfer machines in American automobile factories and members of the staff of Vauxhall Motors, Ltd., on visits to the United States had been greatly impressed by the results obtained through the use of such machines. So much so, in fact, that a decision was made to use transfer machines in the Luton factory. Through the co-operation of James Archdale & Co., Ltd., Birmingham, two transfer machines have been developed for the machining line for the improved cylinder head for Bedford truck engines. They have now been in productive use for sufficient time to prove their efficiency, in fact, the results have exceeded expectations.

Machining sequence

Specially designed Archdale vertical milling machines have been installed

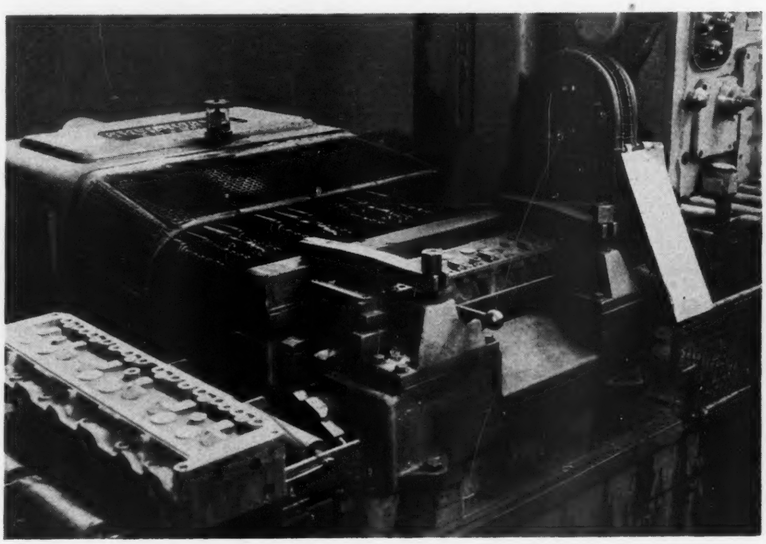


Fig. 4. An Archdale combined drilling and milling machine specially developed for Vauxhall Motors Ltd.

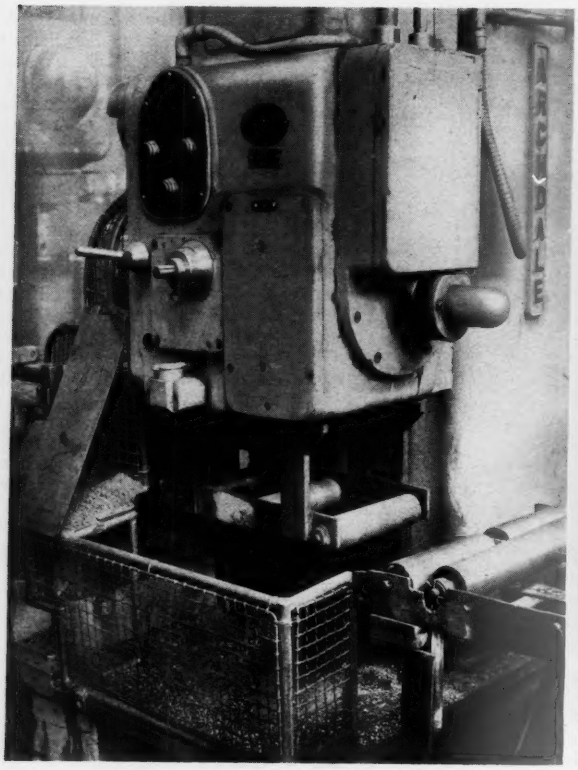


Fig. 5. The milling head of the machine shown in Fig. 4.

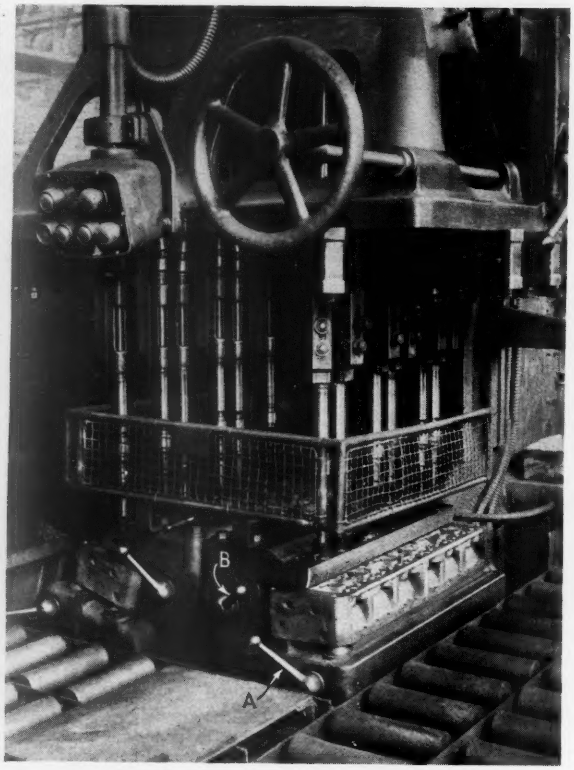


Fig. 6. Set-up for reaming the inlet and exhaust valve guide holes.

for the first and second operations on the cylinder head. On the first machine the rocker face is milled, and on the second, the joint face. A considerable amount of stock has to be removed from each face and a high rate of stock removal is essential to allow these two machines to keep pace with the rest of the line. In addition, the depth of the head must be maintained within close limits and, since the succeeding operations are carried out in some cases with location from the rocker face and in others from the joint fact, a close approach to absolute parallelism must also be maintained.

The special Archdale vertical milling machines used for these operations are illustrated in Fig. 1. Because of the high rate of stock removal and the accuracy that must be maintained these machines are very highly powered, each has a 50 h.p. motor, and the design, particularly of the spindle and the table, gives great rigidity and freedom from chatter. They can employ a table feed as high as 39in per minute, but they are being used at 26in table feed. The Ardalloy cutter has 20 teeth with an effective diameter of 10in. It runs at 100 r.p.m. to give a surface cutting speed of 300ft per minute. The teeth have 10 deg negative radial rake, 10 deg positive axial rake and 20 deg approach angle. Because of the high rate of metal removal there is no heat transfer from the chip to the work, and consequently there is complete freedom from heat distortion.

The rocker face is finish milled from the rough in one pass through the first machine. It would also be possible to finish mill the joint face in one pass on the second machine, but it has been thought advisable to leave a finishing allowance on this face. This face is finished to size at the final operation in the machining sequence to ensure that any accidental damage that may occur

Fig. 7. The loading end of the transfer rack on a transfer machine.

is removed by the finishing cut.

One operator works both machines. The simple, open type of fixture that is employed, see Fig. 2, allows the component to be loaded and unloaded with the minimum of effort and time. The first machine is loaded from the front of the line and on completion of the traverse through the machine, the table stops in the back position for unloading. As the second machine is arranged for loading from the back of the line, transfer from one machine to the other is easily effected and when the joint face is machined the unloading position is adjacent to the ma'n roller conveyor track.

At the next operation both side faces are milled simultaneously on a Cincinatti duplex miller. The new equipment for this operation is not yet available. Therefore it affords a comparison between the old methods and the new. The difference lies not in the actual machining operation but in the method of loading and unloading. A two-tier fixture is used so that two heads can be machined in one pass. This is a departure from the principle of a standard loading height in relation to the roller track. As a result the castings must be lifted in to and out of the fixture. This causes considerable operational fatigue.

From the duplex miller the head is transferred to a Baush multi-spindle drilling machine on which the bolt holes are drilled half through. At the next operation important locating

points, which are used at every succeeding operation, are produced by reaming four corner holes on a Herbert 4-speed sensitive drill. This is a three-spindle machine, but only the outer heads, each of which carries a two-spindle head, are used. The set-up for this operation is illustrated in Fig. 3. Once again attention may be drawn to the simple open type of fixture that is employed. It incorporates air clamping. After reaming, the component is transferred to a second Baush multi-spindle drilling machine on which the bolt holes are finish drilled through from the rocker face and five holes are drilled for rocker studs.

The next machine in the line is an 18-spindle Archdale adjustable multi-drilling machine. In effect two separate operations are completed at every traverse of the machine, seven rocker bracket stud holes are drilled in one face and ten holes are drilled in the tappet cover face. To allow two different faces to be drilled at one traverse, two fixtures are necessary, one for the rocker bracket stud holes at the front of the machine table and the other at the back of the table for the holes in the tappet cover face. The machine is arranged at right angles to the line and loading is effected from the side into open-ended fixtures.

A similar machine is used for the next operations, drilling and reaming the holes for the push-rod tubes. This machine is also at right angles to the line. The holes are combination drilled and reamed from one side, and the head is then turned over in the same fixture for combination drilling and reaming from the other face. It is in order that accuracy shall be maintained in operations such as this that great care is taken to ensure that the faces milled at the first two operations shall be parallel. Further combination drilling and reaming operations are then carried out on an Archdale 20-spindle multi-drilling machine.

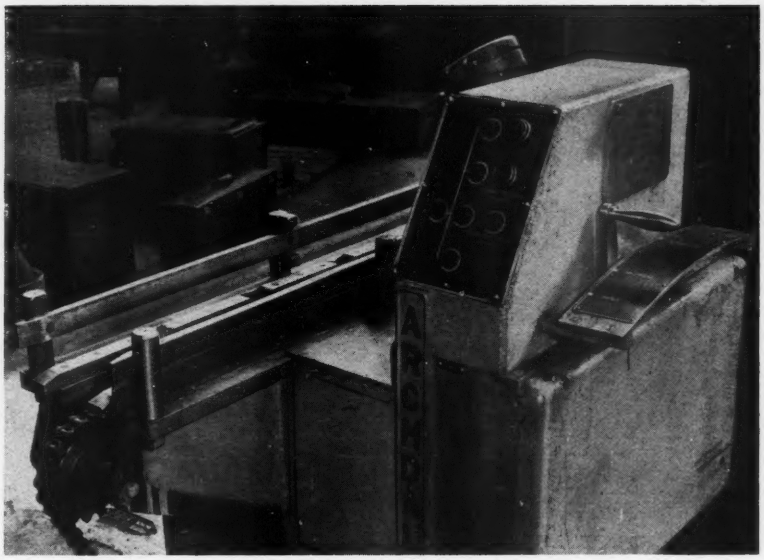


Fig. 8. Control box for a transfer machine.

At this stage the component is transferred to a combined drilling and milling machine that has been specially developed by J. Archdale & Co., Ltd., for use in this cylinder head line. It comprises an 11-spindle drilling head with horizontal traverse and a milling head with vertical traverse. This machine is shown in Fig. 4. The drilling head carries eight twist drills for drilling manifold stud holes and three combined core drills and counterboring tools for machining three induction ports. To ensure accuracy of depth at counterboring there is a timed dwell at the end of the drilling head advance. The milling head is used to machine the water pump facing on the front end of the component.

This is the first operation at which the work-holding fixture is in direct line with the roller track. Consequently, the operator merely has to slide the cylinder head along the track and into the fixture against a stop that gives approximate location. Accurate location is effected by register of two plungers in two of the reamed holes. A single lever actuates both plungers. Until they have registered in the holes and the clamps are down, the machine cannot be started.

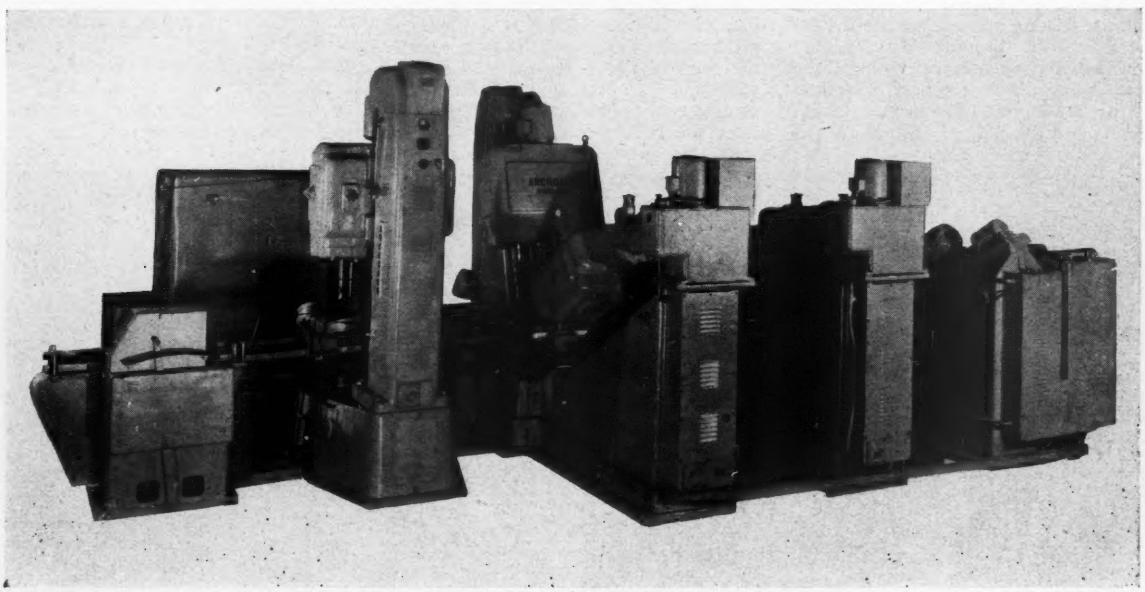


Fig. 9. No. I transfer machine.

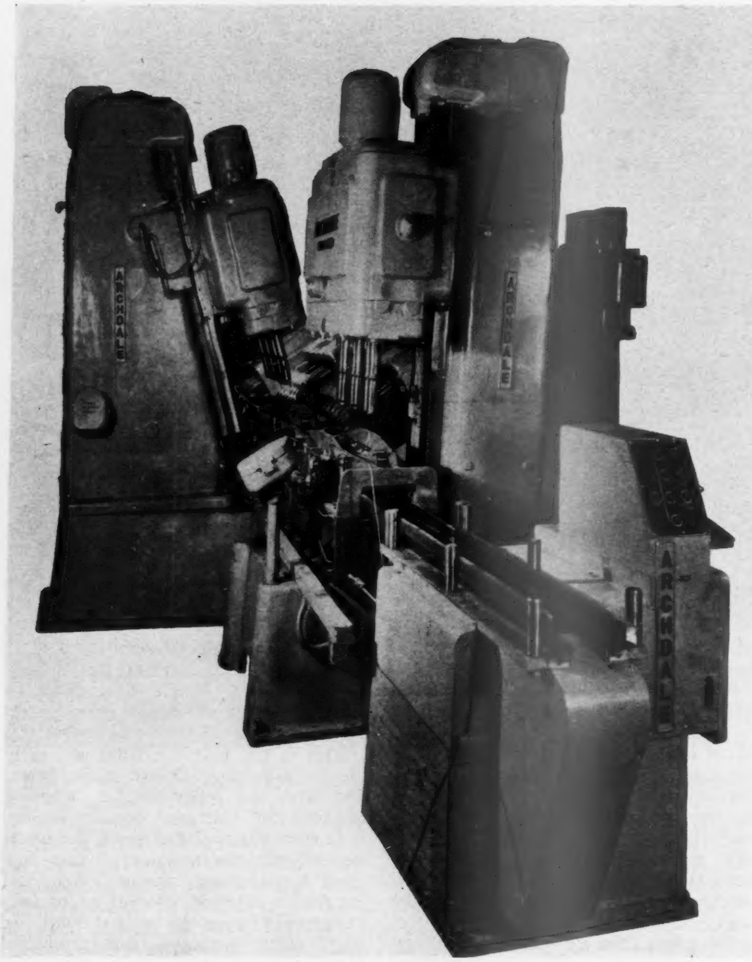


Fig. 10. No. 1 transfer machine from the loading end.

When the machining is completed and the drilling and milling heads have retracted, the locating plungers are withdrawn, the clamps released and the casting is pushed out at the far end of the fixture on to the roller track for transfer to the next machine. However, as the milling head works in the line of the track, there is a distinct gap between the end of the fixture and the start of the next length of roller conveyor. Therefore, to allow the component to be transferred easily from the fixture to the track a short length of roller track is mounted on the underside of the milling head (see Fig. 5). It is in such a position that when the head is fully retracted it lines up with the work-holding fixture and the roller track.

There is, of course, a possibility that the operator may push the casting clear of the work fixture but not clear of the milling head. If this were to happen, considerable damage would be caused to the machine in the next cycle since the milling head would foul the incompletely cleared casting. To avoid any possibility of this, a spring-loaded roller is incorporated in the length of track immediately clear of the milling head. If this roller is depressed, as it must be if the component is pushed clear of the fixture but not clear of the milling head, a limit switch is operated and the starting circuit is broken, so that the machine cannot be started.

From the combined drilling and milling machine the casting passes to an Archdale 12-spindle multi-drill and thence to another Archdale 12-spindle multi-drill. On the first of these machines the valve guide holes are drilled and on the second they are semi-finish reamed. Except for the necessary difference in tools, the set-up is the same on each machine. Two fixtures are mounted on each machine table. The inlet valve guide holes are drilled or reamed in the first fixture and the exhaust valve guide holes in the second. It is necessary to use two fixtures because the inlet holes and the exhaust holes are in different planes.

As with the other two-fixture multi-spindle drilling operations, open-ended fixtures are used and loading is effected from the side of the machine. The type of fixture employed is shown in Fig. 6. The component is slid into the fixture up to a stop, the locating plungers are actuated by lever A, and the clamping is effected by movement of lever B. Neither machine can be started until the location and clamping are effected.

Incidentally, Fig. 6 shows the set-up for reaming the valve guide holes. At this machine, loading and unloading the first fixture are effected from the

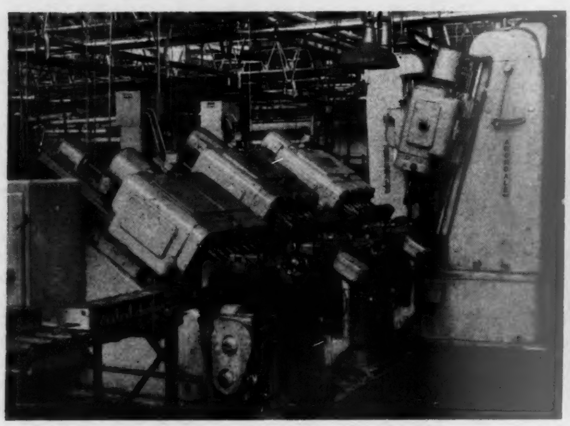


Fig. 11. No. 1 transfer machine from unloading end.

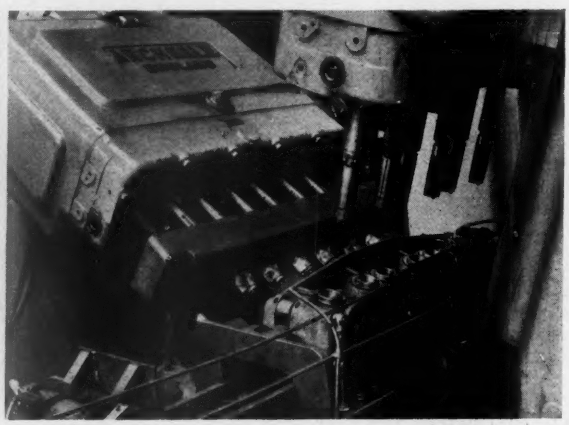


Fig. 12. Second working station on No. 1 transfer machine.

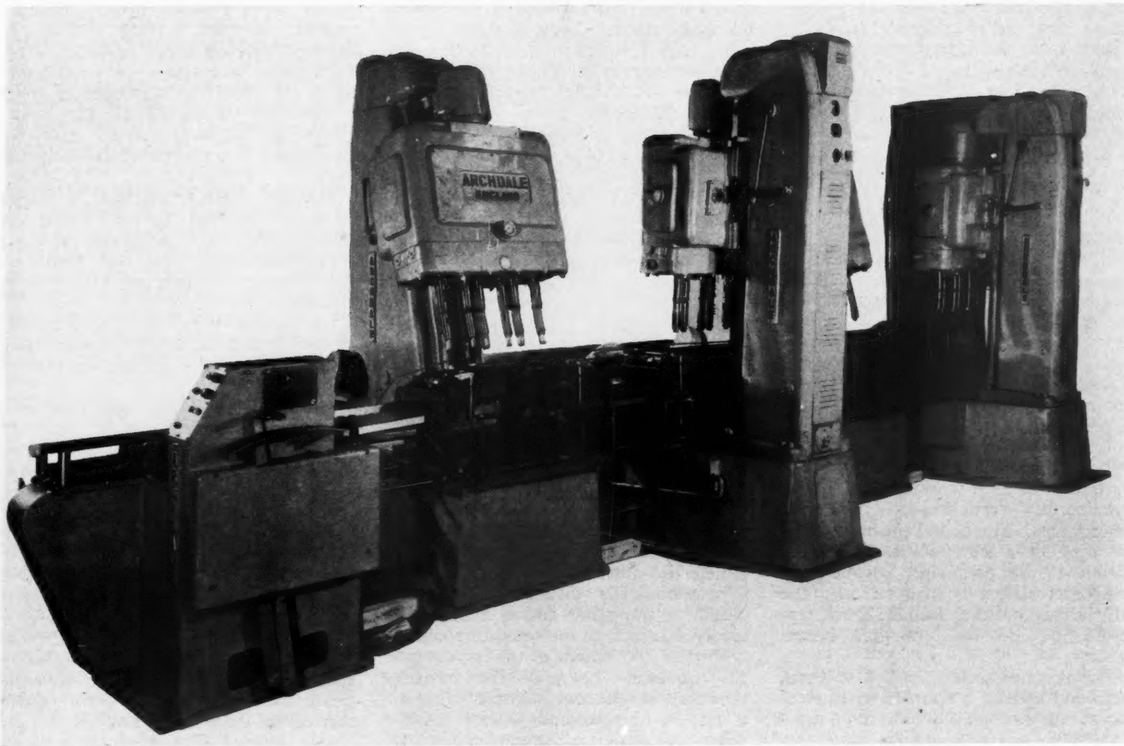


Fig. 13. No. 2 transfer machine.

Fig. 14. No. 2 transfer machine from the loading end.

Fig. 15. No. 2 transfer machine from unloading end.

CYLINDER HEAD PRODUCTION **(*Continued*)**

side of the machine. For the second fixture, that at the right, loading is effected from the side of the machine, but when the reaming is completed the cylinder head is withdrawn from the front. This allows direct movement of the component to the first of the transfer machines.

Transfer machines

As the second transfer machine is immediately behind the first transfer machine and the same type of transfer mechanism is used for both machines, it will be convenient to describe the mechanism before describing the various functions performed on these machines. Each machine has four working stations. Running the full length of the transfer rack system are two steel strips, one on either side of the rack, on which the casting rests. These strips form the base of the work-holding fixture at each working station and give correct vertical location of the components. The actual pusher system for moving a component from one station to the next runs between the horizontal strips. Also running the full length of the transfer rack are machined side rails to give lateral location.

The transfer rack is in five sections. Each end section is coupled to an electric motor through a clutch and a driving chain, and each intermediate section is coupled to the section before and behind it through tie-rods. The loading section for one of the machines is shown in Fig. 7. At each station there is a pusher for carrying the block forward. In Fig. 7 the pusher for the loading section is shown at A. It is so designed that it rolls forward to allow the component to pass over it, and as soon as the head has passed completely over it returns to the working position. Similar pushers are mounted in the other four sections of the transfer rack system. Each of these, however, rolls forward to pass under the appropriate cylinder head on the return stroke of the transfer rack.

At each of the working stations of the transfer machine there is a stop to give approximate longitudinal location of the block. Each stop is arranged to pivot to allow the block to pass over it for transference to the next station. To bring the stops back into position there is an actuating pin in each section of the transfer rack. The pin in the loading section is shown at B in Fig. 7.

All the motions for all four working stations of the machine are controlled through a single quadrant-mounted lever (see Fig. 8). To start the machine cycle the lever is moved from the neutral position to the index position. This causes the clutch at the entrance end to engage and the rack system is moved forward an amount corresponding to the distance between working stations. When this movement is completed the clutch at the entrance end is automatically disengaged and the clutch at the other end is engaged to return the rack to the starting position.

Further movement of the lever round the quadrant operates the locating plungers at all four stations through a system of rods and levers. Only when the plungers are correctly registered in the cylinder heads is it possible to move the lever still further round the quadrant. This further advance of the lever opens valves for air-operated clamps at each of the working stations. This clamping must be effectively completed before the lever can be moved to the final position to start the motors for the various heads.

Each of the heads has an independent and completely automatic cycle. That is, the spindles start, the head advances for a predetermined distance, retracts, and the spindle stops. A series of warning lamps is mounted on the control box and as a head completes its cycle the appropriate lamp is lighted. When all the lamps are lighted the cycle is started again by manipulation of the quadrant-mounted lever.

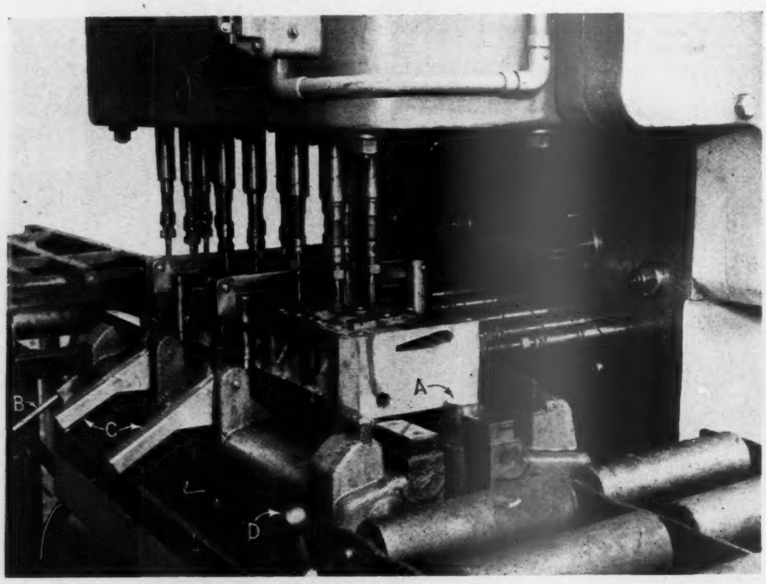


Fig. 16. Set-up on a special Atchdale two-way multi-tapper.

The first transfer machine, which is illustrated in Figs. 9, 10, 11 and 12, incorporates four 6-spindle angular multi-head drills and one 6-spindle multi-tapper. They are arranged as follows:—

1st station.—One drill head tooled for spot facing the top of the inlet valve guide holes. It has an auto-timed dwell.

2nd station.—There are two drill heads, one on each side of the work fixture. One head with auto-timed dwell spot faces the exhaust valve guide holes. The other is used for drilling the spark plug holes. It is not possible to incorporate pilot bushes for the drills in the work fixture, so a casting in which the pilot bushes are carried is spring-mounted on pillars attached to the machine head (see Fig. 12).

3rd station.—One drill head with combination tools for counterboring spark holes. This also has an auto-timed dwell and a pilot bush casting carried on the drilling head.

4th station.—One multi-head tapper for tapping the spark plug holes.

On the second transfer machine there are also four working stations, but on this machine there is only one head at each station. All the heads have an auto-timed dwell. This machine is illustrated in Figs. 13, 14 and 15. The functions carried out are:—

1st station.—Core drill exhaust valve pockets.

2nd station.—Core drill and face an angle on the inlet valve pockets. Combination core drill and facing tools are used.

3rd station.—Semi - finish exhaust valve seat bores to depth and face an angle. A finishing allowance is left for removal at a later fine-boring operation.

4th station.—Semi-finish inlet valve pockets.

These two machines represent what is probably the most advanced production technique employed in the British automobile industry. Their development is the result of close co-operation between Vauxhall Motors, Ltd., and J. Archdale & Co., Ltd. Each machine calls for an intricate combination of mechanically, pneumatically and electrically controlled movements that are closely interlocked and must work with uniform precision. Perhaps the best illustration of what these machines mean in terms of productivity is that, although each machine has four functions, the operators look upon them as single machines. In fact, a single operator tends the first transfer machine and the three machines immediately preceding it, and admits that he is less fatigued at the end of the day than he used to be when working a single machine.

From the second transfer machine the casting passes to another machine that has been specially developed for this line. It is an Archdale 2-way multi-tapper on which 14 holes are tapped on the top face and eight on the manifold face. The machine is so placed that the work-holding fixture is in line with the roller track, see Fig. 16. Incidentally, the work fixture for this operation is a good example of clever design to produce a very simple, yet extremely efficient, method of locating and holding the work. In practice the casting is pushed forward from the roller track until it contacts the stop A. The locating plungers are then brought into position through movement of the lever B and the clamps are actuated through simple movement of the levers CC. So that the work can be pushed forward on to the next length of roller track, the stop A is retractable by movement of the lever D.

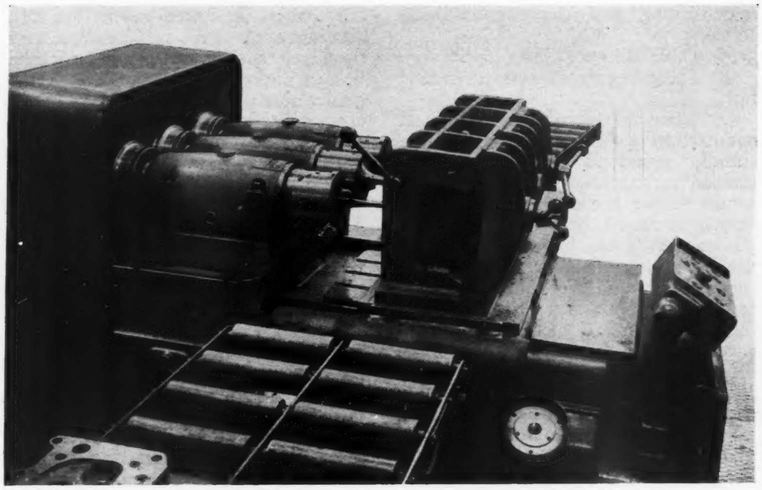


Fig. 17. Precimax fine borer for finishing inlet valve guide holes and valve seats.

At present the component moves forward from the tapping machine to three sensitive radial drills. As these machines will not be used when retooling is complete there is no point in describing them. At this stage every head is inspected. After inspection and before any further machining, various assembly operations are carried out, and the head is given a water test and is cleaned, before the valve guide holes and the valve guide seats are finish machined.

On any component that passes through a multiplicity of machining operations there is usually one operation that gives rise to more necessity for rectifications, if not outright rejections, than any other. With cylinder heads it has been the experience of Vauxhall Motors Ltd., and we believe this is true of other organizations, that the greatest single cause of trouble was lack of concentricity between the valve guide bore and the valve seat. In the re-tooling of this line, the machining methods in connection with the valve elements have been radically altered to give improved results.

Previously, it was the practice to finish machine the valve seats after the valve guides had been inserted in the cylinder head. This method of machining the seat from the valve guide bore appears to be the logical one, but in practice a fair amount of rectification was necessary to maintain the prescribed standard of concentricity. Now, there has been a complete break with conventional practice, and the machining sequence is such that the valve seat is fully machined before the valve guides are inserted; the guides are also finish machined before insertion. The results have proved that the change was fully justified.

For machining the valve elements in the cylinder head special Precimax fine boring machines, see Fig. 17, have been developed by John Lund and Co. Ltd., Crosshills, Keighley. There are two of these machines in the line. One is for machining the inlet valve elements and the other for the exhaust valves. Apart from the necessary differences in tool setting, the machines are identical.

Each machine has three special heads so that two passes are required for machining the elements for all six valves. This cannot be avoided, because although the machine heads are marvellously compact, the minimum possible centre distance between heads is much greater than the centre distance of adjacent valves. Each head carries three tool bits, one for boring the hole to take the valve guide, one for turning the valve seat and one for machining a counterbore. The boring and counterboring tools are fixed in relation to the spindle axis, but the turning tool for the valve seat operates from a taper slide in the head to produce the correct valve seat angle, see Fig. 18.

This Precimax machine has a complex but fully automatic cycle for machining all six valve seats and holes for the valve guides. The casting is loaded with Nos. 1, 3 and 5 positions in line with the machine spindles and the following sequence is carried out:

(1) Rapid advance of table.

(2) Table advance reduced to feed rate.

(3) Holes for valve guides are bored.

(4) When the boring tools are clear the spindle speed is automatically reduced so that the correct surface cutting speed is maintained for the valve

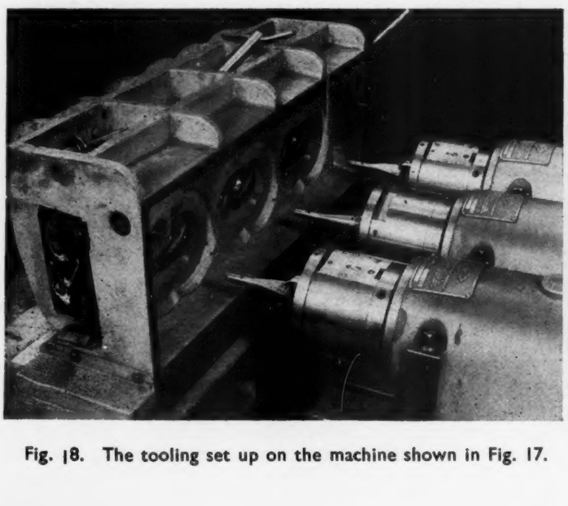


Fig. 18. The tooling set up on the machine shown in Fig. 17.

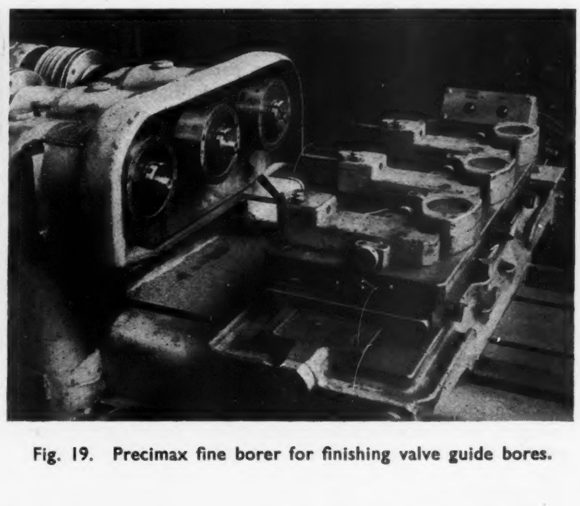


Fig. 19. Precimax fine borer for finishing valve guide bores.

CYLINDER HEAD PRODUCTION (Continued)

seat turning and the counterboring tools.

(5) Table advance completed.

(6) Table begins to retract at feed rate.

(7) When the valve seat turning and counterboring tools are clear the spindle speed is automatically increased to the original rate. On the return stroke the boring tool takes a very light cut on the bore for the valve guides. This tool shows a slight degree of spring on the relatively heavy forward cut, and the light cut on the return stroke is important in maintaining the necessary close dimensional accuracy and for producing a high standard of surface finish.

(8) When all tools are clear there is rapid table traverse to the out position.

(9) Table indexes to bring No. 2, 4 and 6 into line with the spindles.

Movements 1-8 are then repeated and finally the table indexes back to the starting position. This sequence applies to both machines.

Although the machine for boring the valve guides is not included in the cylinder head line, brief reference may be made to it since full advantage of the Precimax machines can be taken only when the valve guides are also accurate. A three-spindle Precimax machine, see Fig. 19, is used for fine boring the valve guides. Before the guides reach this machine, the O.D. is accurately finished and the bore is rough drilled and reamed to leave a finishing allowance.

The guides are mounted in accurate double diaphragm chucks in the machine heads, and the tools are carried in holders mounted on the machine table. The table advances with the tools clear of the work, and at the end of table advance each tool has passed completely through a valve guide. At this point the table is automatically indexed an amount that brings the tools into the cutting positions and boring is carried out during the return stroke. At the completion of the return stroke the table indexes to the starting position and the machine stops. It has been found that this method is much more accurate than the previous method of finishing the bore by reaming.

There is, of course, a very slight degree of closing in when the valve guide is inserted into the cylinder head since it is an interference fit. Therefore to ensure correct sizing a reamer is run through every valve guide. Experience has proved that this new method gives greatly improved results, and the need for rectifications because of lack of concentricity between the valve guide bore and the valve seat are virtually eliminated. At this stage the machining is complete except for the final operation at which a finish milling cut is taken on the joint face.

Although this line is not yet fully re-tooled, the results already obtained are remarkable. The total cycle time for a cylinder head has been reduced from approximately 120 minutes to approximately 80 minutes. That is, the productivity of the line has been increased by rather more than 30 per cent. This, in itself is an achievement, but in addition there has been a much greater than 30 per cent increase in the productivity per man hour. Actually the new line employs nine fewer machinists than were required on the old line, but because of the greater complexity of the machines an additional tool-setter is required. There is therefore, a total saving of eight men and the productivity per man hour is increased by something in the order of 70 per cent.

INSTITUTION OF MECHANICAL ENGINEERS

Summer Meeting to be Held in Birmingham

The Summer Meeting of the Institution of Mechanical Engineers will be held in Birmingham beginning on Wednesday, June 21st and ending on Friday, June 23rd. The provisional programme is as follows:—

June 21st.—Morning: Opening meeting, followed by the reading and discussion of a paper. Afternoon: Alternative visits to works. Evening: Reception.

June 22nd.—Morning: Reading and discussion of a paper on "Automobile Test Rigs" by Mr. L. H. Dawtrey, M.I.Mech. E., in Coventry, or alternative visits to works. Afternoon: Alternative visits to works. Evening: Institution Dinner in Grand Hotel, Birmingham.

June 23rd.—Alternative whole-day visits to works or a pleasure excursion. Evening: Conversazione in the University of Birmingham, by invitation of the Midland Branch Committee.

REED VIBROMETER

THE reed vibrometer made by the General Electric Co., Ltd., Magnet House, Kingsway, London, W.C.2, has a reed in the form of a steel strip, the effective length of which can be varied by means of a knob on the side of the vibrometer. When the natural frequency of the reed coincides with the frequency of the disturbance, a scale on the vibrometer indicates the fundamental frequency of vibration. By varying the position of the vibrometer on the structure under test, not only can the natural frequency and modes of vibration be observed, but also the direction in which the maximum disturbance is acting is shown by the maximum movement of the reed. A feature of the G.E.C. vibrometer is that it contains an electro-magnetic pick-up in close proximity to the reed, and it is thus possible to display the results on a cathode ray tube. (1890)

ENGINEERING EQUIPMENT USERS' ASSOCIATION

FIVE companies in the process industry field, namely, the Anglo-Iranian Oil Co., Ltd., Courtaulds, Ltd., Imperial Chemical Industries, Ltd., Lever Brothers and Unilever, Ltd., and the Shell Petroleum Company, Ltd., have formed an Association entitled the Engineering Equipment Users' Association. It will provide its members with a means for exchanging information on standardization of engineering materials and equipment which they use in common. It will work through the British Standards Institution on matters of general standardization, and when experience has been gained it is hoped that arrangements can be made for other user firms to take part in consultations on related matters of common interest. (1886)

TAPER-ROLLER BEARING STANDARDS

FOLLOWING publication of a revision of B.S.292, "Dimensions of Ball Bearings and Parallel-roller Bearings," the British Standards Institution has decided to begin work on taper-roller bearings. As with other bearings, inch and metric sizes are in use in this country, and it was agreed that in revising B.S.292 metric sizes only of ball and parallel-roller journal bearings should be recommended for use in new designs. In the case of taper-roller bearings, however, the predominant usage has been reported to be in inch sizes.

A sub-committee has accordingly been set up to examine the position, and as a first step they have been asked to try to draft a systematic series of standards in the inch system in which the overall dimensions progress by regularly increasing steps. At the moment the work of this sub-committee is purely exploratory. (1891)

WROUGHT STEELS

BRITISH Standards Institution, 24/28, Victoria Street, London, S.W.1, have published B.S.971: 1950, Commentary on British Standard Wrought Steels, En series. It supersedes B.S.971: 1942 and is complementary to B.S.970: 1947, giving guidance on the use of that standard. Emphasis is placed on the mass effect, and there is useful information on the effects of ruling section and heat treatment on mechanical properties. Results of individual investigations covering the range of En steels are included. (1889)

TRACKED VEHICLES

An Analysis of the Factors Involved in Steering

By W. Steeds, O.B.E., B.Sc., M.I.Mech.E.

(*Continued from page* 148.)

Regenerative and non-regenerative steering

IT has been seen that in most cases the longitudinal force acting on the inner track is negative, that is, its sense is fowards and so the inner track is driving its sprocket. This forwards force is associated with a forwards rotation of the sprocket so that the inner track is putting power into its sprocket. In a regenerative steering system this power can be transferred through the system to the outer sprocket and there help to supply the power required by that sprocket; the engine would then have to supply only the numerical difference between the sprocket powers. In a non-regenerative system the inner sprocket power cannot be transferred to the outer sprocket and has to be absorbed in a brake while the engine has to supply the whole of the power required by the outer track and, in one case, more than that.

The powers involved can be surprisingly large. Thus, taking the simple example used in the section on kinematics above where $a_o = -a_i = 1 \cdot 925$ft, suppose that the sprocket speed ratio is $x = \omega_o/\omega_i = 1 \cdot 3$, the mean speed is $V = 10$ ft/sec. and the sprocket radius is $r = 1 \cdot 058$ft. Then the radius of turn is given by equation (12) as 44ft, and equation (13) gives $\omega_i = 8 \cdot 23$ radians/sec. so that $\omega_o = 10 \cdot 7$ rads./sec. Then since $a/l = 0 \cdot 35$ the value of $f(a/l)$ is, from Fig. 6, $0 \cdot 62$ and taking W as $15 \cdot 3$ tons and μ as $0 \cdot 7$ the track forces are $\pm 0 \cdot 7 \times 15 \cdot 3 \times 0 \cdot 62 = \pm 6 \cdot 65$ tons.

Hence the sprocket powers are:—

$$HP_o = 6 \cdot 65 \times 2240 \times 1 \cdot 058 \times 10 \cdot 7/550 = 306.6$$

and $HP_i = -6 \cdot 65 \times 2240 \times 1 \cdot 058 \times 8 \cdot 23/550 = -235 \cdot 8$.

With a regenerative system the engine would have to supply only $306.6 - 235 \cdot 8 = 70 \cdot 8$ h.p., whereas with non-regenerative steering it would have to supply $306 \cdot 6$ h.p.

The systems shown in Figs. 17 and 18 below are non-regenerative, all the others are regenerative. Assuming that no frictional losses occur in the steering mechanisms and that the brakes and clutches, by means of which steering is effected, do not slip then there is no difference between the various regenerative systems so far as the powers are concerned and the advantages of one system over another are in such matters as simplicity of construction, number of different turning radii provided, space occupied and cost of production, etc.

An important division in all steering systems is between the basically continuous systems in which the application of a single brake or the engagement of a single clutch effects the steering and there is no interruption of the drive to the sprockets and the basically discontinuous systems in which two separate operations are needed to initiate a turn, one being the disconnection of one drive and the other the engagement of some alternative drive or the application of a brake.

The discontinuous system always involves the risk of "reversed steering" (i.e., the controls are operated so as to initiate a right-hand turn but the vehicle turns to the left or vice-versa) if the vehicle at the moment the steering is actuated happens to be over-running the engine. Reversed steering can be obviated by arranging for the two operations involved in steering to overlap but to obtain this requires careful adjustment of the controls and may introduce excessive losses. Thus a discontinuous system is undoubtedly inferior to a continuous one, other factors being the same. The systems shown in Figs. 18, 19, 28 and 35 below are discontinuous systems, all the others being continuous.

Steering mechanisms

Most of the important mechanisms that have been proposed and used are shown diagrammatically in Figs. 17 to 35; these will now be briefly described and some indication of the dates at which the mechanisms were first introduced will be given, but while it is believed that these dates are accurate the author does not claim to have made any very exhaustive search in establishing them. The braked differential system (Fig. 17) was the earliest system having certainly been used in 1904 and very likely earlier. Steering is effected by applying the brake on one side, thereby slowing down the track on that side and, assuming the engine speed to be unchanged, speeding up the other side. The system is extremely simple but, as will be seen when the power losses that occur in it when the brake is slipping have been examined, is otherwise a bad system.

Clutch-brake steering shown in Fig. 18 is also one of the very early systems having been used in 1911. Steering is effected by disengaging the clutch on one side and applying the brake on that side. In the early designs the clutch and brake were entirely separate mechanisms but the Rackham clutch, which was invented about 1920, combined them and used the brake to give a servo opera-

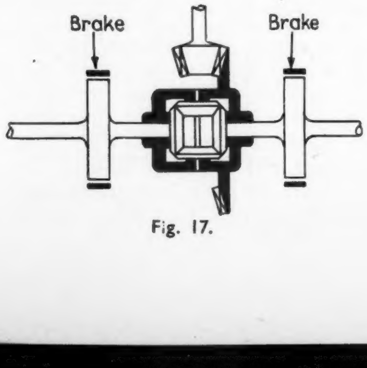


Fig. 17.

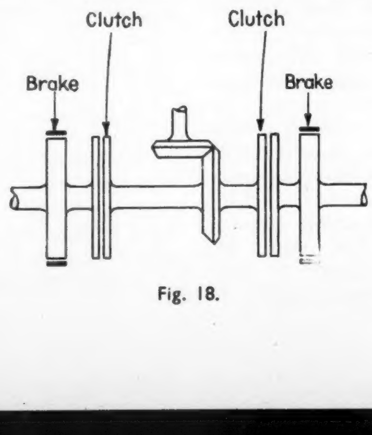


Fig. 18.

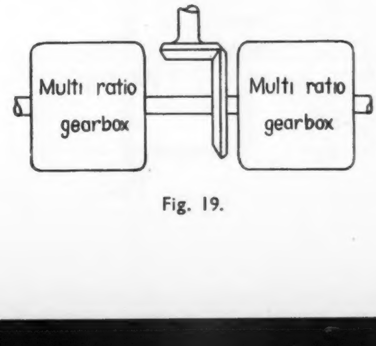


Fig. 19.

TRACKED VEHICLES (Continued)

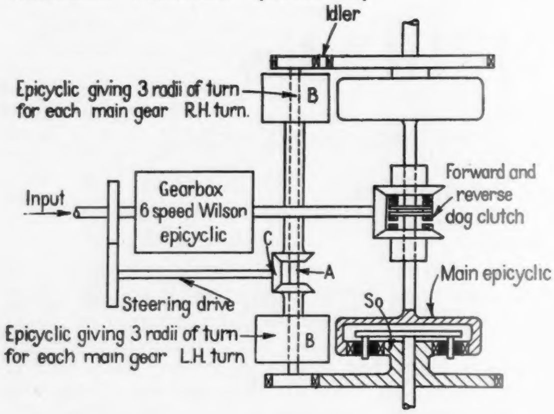


Fig. 20.

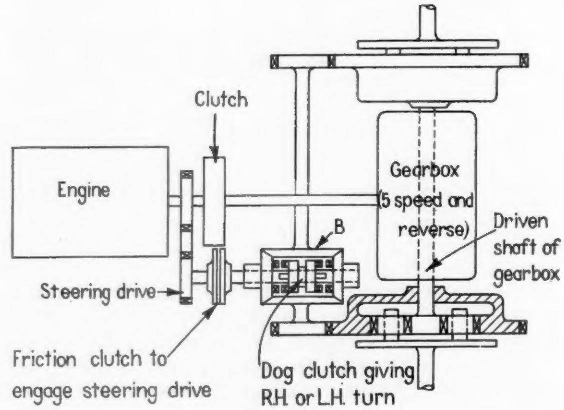


Fig. 21.

tion of the clutch followed by a braking action on the sprocket. Thus only one control was needed and the two actions were practically simultaneous, thus eliminating the risk of reversed steering.

Geared steering was first used in the first tanks designed by Maj. Wilson during the 1914-18 war. These tanks incorporated a two-speed gear in the drive to each sprocket. The system was next used in a greatly refined form in an experimental tank, the transmission system of which was also designed by Maj. Wilson round about 1926-28. In this latter design, indicated in Fig. 19, two six-speed epicyclic gearboxes were used, one in the drive to each sprocket. In a later design two eight-speed boxes were used. Steering is effected by changing down to a lower ratio on one side than on the other side. With the multi-ratio boxes interlocks were provided so that the gear ratio engaged on the inner side could not be more than two or three steps below that engaged on the outer side. This gave two or three radii of turn for each gear ratio provided by the gearboxes thus giving, with the eight-speed boxes, eighteen radii of geared turns and one skid turn. The system, also with Wilson type gearboxes, was fitted to some French tanks.

Another steering system designed by Maj. Wilson round about 1928 but never actually incorporated in a tank is that shown in Fig. 20. It may be called a controlled epicyclic system. The sun gears S_0 and S_1 of the main epicyclics are coupled by shaft A which is geared directly to one sun and through an idler to the other sun. The annuli of the epicyclics are driven through the double bevel drive, giving forward and reverse, from the main gearbox. Thus when the shaft A is left free it serves to balance the torques acting on the sun gears of the epicyclics and thus to balance the sprocket torques. Steering is effected by positively driving the shaft A forwards or backwards according to the hand of turn required. This is done by the steering epicyclics BB whose driving members are coupled by the bevel cluster C to the input end of the main gearbox. The steering epicyclics were arranged to give three ratios so that three radii of turn would be available for each ratio provided by the main gearbox.

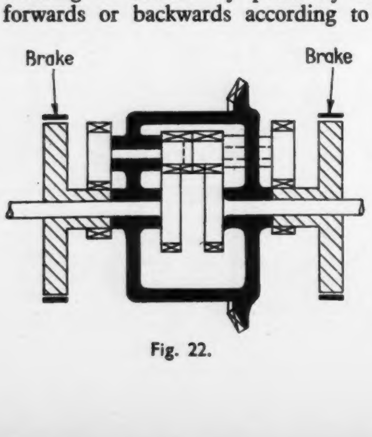


Fig. 22.

A design on rather similar principles by the Rolls-Royce company is shown in Fig. 21. This was fitted to some experimental tanks but never got into production. In this arrangement the annuli of the main epicyclics are the controlled element; they are coupled through the bevel gear cluster B so that when the steering drive is not engaged the torques on the annuli and hence the sprocket torques are balanced. Steering is effected by engaging the dog-clutch to give the desired hand of turn and then engaging the steering clutch so as to drive one annulus forwards and the other backwards. This arrangement gives only one radius of turn for each ratio in the main gearbox which is now arranged between the epicyclics and which drives the shaft carrying the sun gears of the epicyclics. So far as the author's knowledge goes this system was designed about 1940.

The controlled differential system shown in Fig. 22 was first produced round about 1927 and was first fitted to the Cletrac tractors. It has since been very widely used having been adopted for practically all the war-time tanks and tracked vehicles built by the United States and having been

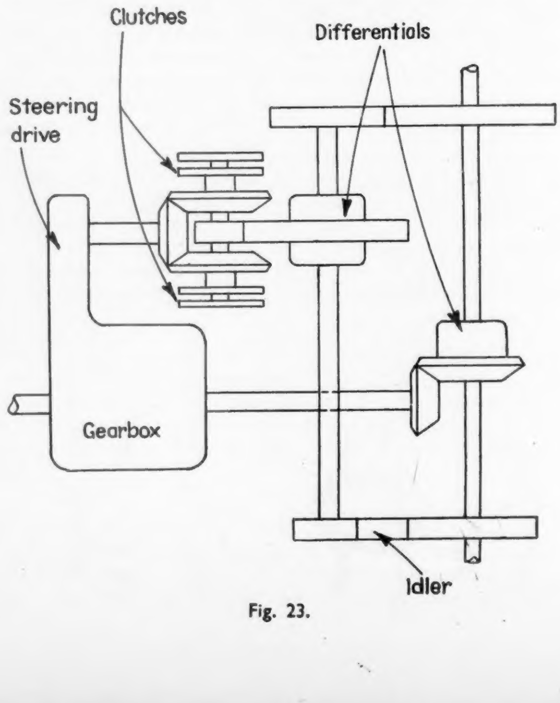


Fig. 23.

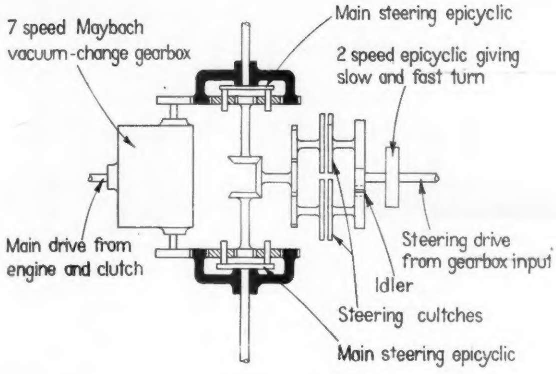


Fig. 24.

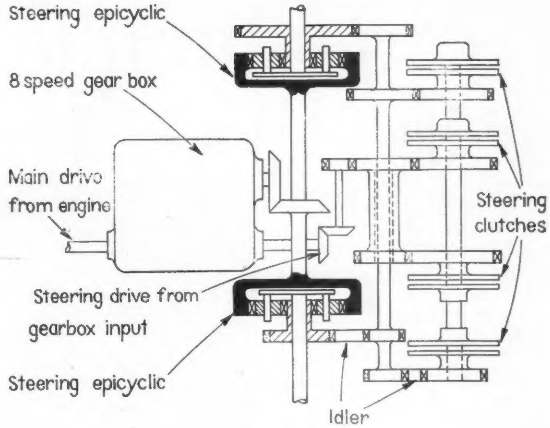


Fig. 25.

used also by most of the other nations that have built tanks with the the exception of ourselves. Steering is effected by applying a brake to one of the controlling sun gears thereby forcing a rotation on the differential pinions and thus producing a difference in speed between the half-shafts. It is of course quite immaterial whether the differential is a spur type as shown or whether it is a bevel type; the latter has been used by the Japanese and, it is believed, by the Russians but it does not appear to be so convenient constructionally as the spur type. The system gives only one radius of turn irrespective of the gear engaged in the main gear-box and this is probably its main drawback.

The double differential system shown in Fig. 23 was used in a French tank built about 1934 but when it was invented is not known to the author. Steering is effected by engaging one of the clutches shown thereby driving the casing of the "steering" differential and producing a difference in speed between the half-shafts of the "main" differential which are coupled to the sprockets. The radius of turn depends on the speed at which the steering differential is driven in relation to the main differential and thus in the form shown gives as many radii as there are gears in the main gearbox.

In the French tank mentioned above the drive to the steering differential was through a variable speed gear of the Hele-Shaw fluid type giving a continuously variable speed, forward or backward and thus providing an infinity of turning radii and "continuous" steering, right through from the smallest possible radius on the one hand to the smallest radius on the other hand with straight running in the middle. Most of the other systems considered have been discontinuous so far as the change from right hand to left hand steering is concerned.

Fig. 24 shows a somewhat simplified diagram of a system designed, about 1938, by Dr. Merritt and his staff for an experimental tank. It will be seen to be identical in principle with Maj. Wilson's design, Fig. 20. The system used in the German "Tiger" tanks and shown in Fig. 25 is also on the same principle.

The well-known Merritt-Brown system is shown (with a four-speed gearbox) in Fig. 26. It has been fully described in several papers including the *Automobile Engineer* (July, 1946) but may be very briefly described here. The annuli of the epicyclics are driven from the differential case member through one of the four pairs of gears while the sun gears are driven, through idlers, by the differential pinions. When the tank is running straight the speeds of the suns are equal and thus the speeds of the sprockets also, the latter being compounded of a forward component from the annuli and a backward component, much smaller in magnitude, from the suns.

Steering is effected by applying the brake to one of the suns. This has the effect, usually, of bringing that sun to rest and consequently of speeding up the sprocket on that side since the backward component has disappeared; on the other side, however, the sun gear will now be driven at twice its previous speed so that the backward component is doubled and the sprocket on that side will run slower than before. Since the component added at one side and subtracted at the other side when one sun gear is fixed is constant for a given engine speed whereas the components due to the annuli depend on the gear ratio engaged between the differential case member and the annuli shaft it follows that there are as many turning radii as there are gear ratios. If no gear is engaged there will be no drive to the sprockets unless one of the steering brakes is applied. In this case the annulus shaft will act as a balance shaft to balance the torques applied to the sprockets and, assuming one sun to be fixed by its brake, one sprocket will, probably, be driven forwards and the other backwards at equal speeds but this depends on the resistances to the motions of the sprockets and the backwards rotating sprocket might rotate faster than the forwards rotating one. If the annulus shaft is locked to the gearbox case by the dog-clutch shown then the forward components of the sprocket speeds will be zero and only the backwards components will remain. Thus a reverse gear is obtained.

The "Twin-turn" system shown in Fig. 27 was developed, about 1941, by the Hydraulic Coupling and Engineering Company. Examination will show that it is identical in principle with Maj. Wilson's 1928 design. When the steering clutches are disengaged the shaft J acts as a balance shaft to balance the sprocket torques, when one of the steering clutches is engaged then one sun gear will be driven forwards and the other backwards thereby producing the necessary difference in speed between the sprockets. Engagement of the other clutch gives the opposite hand of turn. As shown the system gives only one radius of turn.

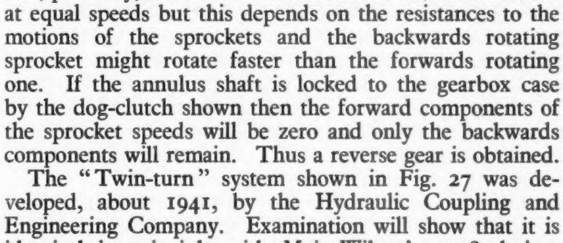


Fig. 26.

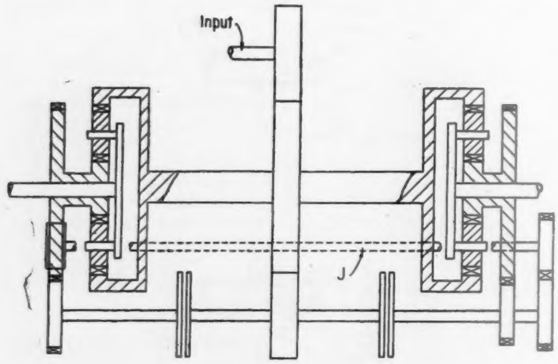


Fig. 27.

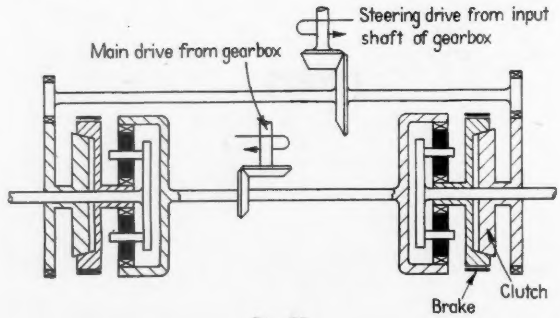


Fig. 28.

The system shown in Fig. 28 is that fitted to the German "Panther" tanks; for straight running both steering clutches are engaged and both brakes are applied so as to fix the sun gears; the epicyclics then give equal reductions and the sprocket speeds are equal. There is, however, no balance of sprocket torques, i.e., there is no differential action and this was claimed as one of the advantages of the system by the Germans. To steer, the brake at one side is released and the clutch at that side is engaged so that the sun at that side is now driven in the opposite direction to the annulus and the sprocket speed at that side is reduced.

In practice the mechanism was adjusted so that the brake and clutch overlapped in order to avoid reversed steering. Since the drive to the sun, when the clutch is engaged, comes from the front of the main gearbox while the drive to the annuli comes from the rear the sprocket speed ratio will depend on the gear ratio engaged in the main gearbox and the system will provide as many turning radii as there are ratios in the gearbox. If the engine speed remains the same after the engagement of the steering clutch as it was before then the mean speed of the tank when turning will be lower than when running straight. This feature, which is common to all forms of geared steering, of which the Panther system is a particular form, enables the momentum of the vehicle to be used to assist in initiating the turn.

The last system to be shown (Fig. 35) is the Gates L.G.1 system. This was developed round about 1942 and has been fully described in the *Automobile Engineer* for February, 1947. When running straight the two clutches at the ends of the input shaft are both engaged and the clutch at the differential case is disengaged. There is thus a positive drive to each sprocket and no balance of sprocket torques. To steer, the differential clutch is engaged and one of the clutches on the input shaft is released. This results in the differential case being driven at a lower speed than the half-shaft on the side where the input shaft clutch is still engaged and so the other half-shaft is driven, via the differential, at a lower speed than the differential case.

Tanks have been built in which a complete power plant and transmission system was provided for each track, the first such system being that used on the Medium C tanks of the 1914-18 war. In view of what has been said about regenerative steering it will be seen that there is not much to be said for this system.

(To be continued)

FRICTION AND LUBRICATION

Investigations on Metal Transfer Effects

STUDIES made by Dr. J. T. Burwell, Jr., associate professor of mechanical engineering at the Massachusetts Institute of Technology, show that traces of metal worn from the piston rings in a gasoline engine can actually be detected forming part of the cylinder walls. Only radioactive "tracing" techniques make possible measurements sensitive enough to show these effects.

The basic method is to make radioactive the atoms in one block of material which is to be rubbed against another in a friction test. After the two are rubbed together, radioactive atoms from block number one appear on the surface of number two, and tests show that they have actually become part of the second block. It is believed that atom transfer probably takes place also in the other direction. Thus it is probable that "untagged" normal atoms from block number two appear on number one after such a friction test. Even with good lubricants between the blocks the same transfer effect is observed, though on a much reduced scale.

The first friction studies were made with two cylindrical specimens, one of which was held stable while another turned against it. After a specimen made radioactive in a cyclotron had rubbed against a normal one for a few rotations, there were bands of radioactivity on the normal specimen. The amount could be counted with a Geiger-Muller counter, but more accurately it could be measured by observing the streaks on a photographic film left wrapped around the cylinder for a week or more. It is reported that the only effective way to remove the radioactivity is by honing, which gives an idea of the strength with which the radioactive atoms have welded to the non-radioactive surface.

A single-cylinder test engine in the Sloan Automotive Laboratory fitted with piston rings plated with radioactive chromium showed metal transfer from ring to cylinder wall during even the mildest conditions of engine operation. These tests were made immediately after installation of new rings, that is to say during the "break-in" period, where a high rate of wear would be expected. Shortly it is hoped to use solid rings of radioactive iron with which it should be possible to study transfer, if any, after the ring and cylinder combination is past the high-wear, run-in period.

According to Professor Burwell the amount of metal left on one block by rubbing another against it may be as little as one-hundred-millionth of an ounce. The chromium transferred from piston ring to cylinder wall in a single-cylinder engine was less than three ten-millionths of an inch thick, but in both cases it is claimed that photographic film gives an accurate picture of the result. It is suggested that radioactivity is the most powerful tool yet used to study metal transfer as a factor involved in friction. (1892)

AUTOMATIC MACHINING

A Spindle Locating Attachment Developed by A. C. Wickman, Ltd.

ON multi-spindle chucking automatics, it frequently occurs that the unloading and loading can be carried out in appreciably less than the total cycle time. This free time at the final station has always been a challenge to the tooling engineers, inasmuch as it does introduce the possibility of an additional operation. An ingenious and interesting attachment for making effective use of this free time has recently been developed by A. C. Wickman, Ltd., Tile Hill, Coventry, for use on Wickman multi-spindle chucking automatics.

This attachment is designed to locate the spindle at the loading station and lock it in position for a portion of each cycle before unloading is effected. It therefore creates conditions in which an operation, such as drilling or milling, can be performed through the medium of an independent live attachment while the component remains stationary. It is also so designed that the ancillary machining operation is carried out in consistently positive and precise relationship with the form of the component.

Immediately after the machine spindles have indexed, the following operational sequence is applied to the attachment. The spindle driving clutch is disengaged and the brake applied to bring the spindle to rest at station number five, as for unloading and loading in the usual manner. Through the operation of a special cam fitted on the standard cam drum, the brake is then released. This leaves the spindle free to revolve. An auxiliary slow-motion drive is then engaged to rotate the spindle until a detent slot is brought into the correct position for engagement with a locating dog. Engagement of the detent slot and the locating dog disengages the slow-motion drive and leaves the spindle stationary in a positively located position. On completion of the ancillary machining operation and after unloading and loading, the spindle clutch is re-engaged in the normal manner in preparation for the start of the next machine cycle.

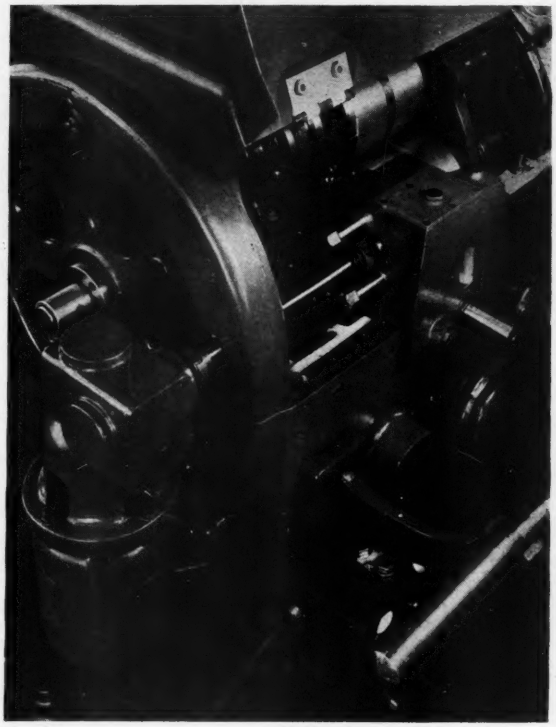


Fig. 1. Drive motor and reduction gear unit for spindle location attachment.

At the rear of each spindle, a ring gear for transmission of the slow-motion drive and locating ring to take the locating dog are together bolted and dowelled to the outer member of the air cylinder which is dowelled in position on the spindle. A fractional horse-power motor is mounted on the index casing of the machine. Through a worm and wheel reduction gear unit and a flexibly coupled shaft, it drives the engaging gear unit of the attachment.

The engaging gear unit is mounted on the side of the index casing in a position adjacent to the air cylinders, as illustrated in Fig. 1. It consists essentially of a cast aluminium housing in which a freely mounted countershaft connects with the flexible coupling drive. A sleeve is carried on the countershaft. Intermittent transmission of the drive to this sleeve is provided by a sliding dog keyed to the countershaft and arranged to slide in and out of engagement. The engaging pinion is keyed to the sleeve. It is arranged to slide longitudinally into mesh with the ring gear mounted on the spindle.

A disc cam governs the action of the mechanism. It is mounted on the end of the existing camshaft. On each side of the disc cam there are masking segments that operate two vertical, spring-loaded bell-crank levers which are located on a bracket inside the wall of the index casing. One bell-crank lever carries a locating dog shaped in the form of a broad wedge. When the lever is released by the cam, this dog rides on the periphery of the locating ring on the spindle. The other bell-crank lever is so arranged that when released by the cam it withdraws a restraining yoke to allow the slow-running pinion to move longitudinally, under spring pressure, into engagement with the ring gear on the spindle.

Through this arrangement the spindle is caused to rotate until the wedge-shaped locating dog engages with the detent slot in the locating ring on the spindle. This further movement of the lever that carries the locating dog, actuates a lever and yoke mechanism to disengage the sliding dog from the sleeve on the countershaft. This disengagement of the sliding dog from the sleeve disconnects the drive to the engaging pinion and leaves the spindle stationary and positively located. Upon conclusion of the machine cycle, and immediately before spindle indexing occurs, the two bell-crank levers are re-set by the action of the disc cam. The return movement of the levers disengages the engaging pinion from the ring gear on the spindle, withdraws the locating dog from

Fig. 2. Flange couplings produced on a Wickman chucking automatic with spindle location attachment.

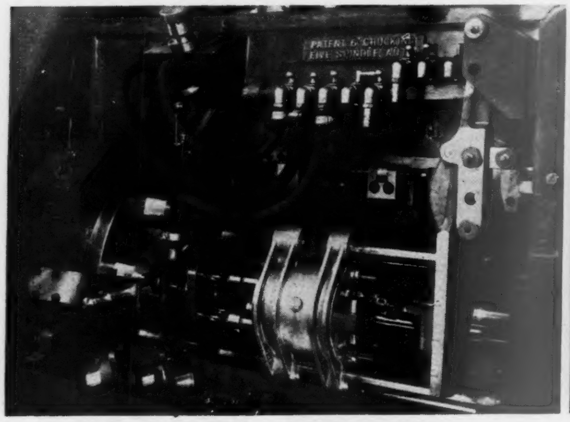


Fig. 3. Multiple drill head at fifth station for drilling holes during free time before unloading.

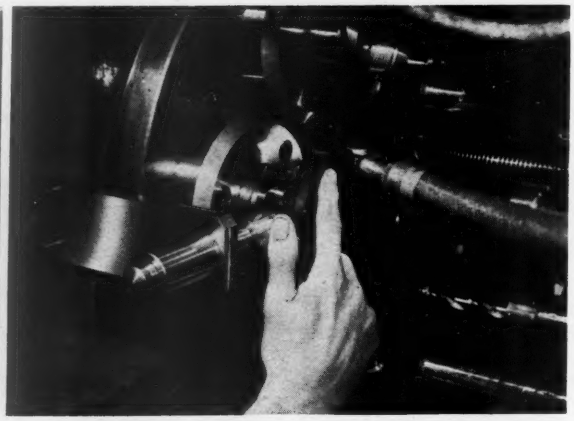


Fig. 4. Loading, the component, which is mounted on a special mandrel, in the correct relative position.

the detent in the locating ring, and causes the sliding dog to re-engage with the sleeve on the countershaft.

A drilling application

This attachment has been used on a Wickman 6in multi-spindle chucking automatic in the production of the flanged couplings for universal joints, illustrated in Fig. 2. On this component, what would otherwise have been free time at the loading station is used for the performance of a multiple drilling operation that would normally have been carried out as a subsequent operation on another machine.

In the production of this component the operations for skimming the long boss, drilling through and broaching are planned to precede the automatic stage. On the automatic, the first four spindle stations are used for rough and finish boring, taper boring, rough and finish facing both sides of the flange, and recess forming the counterbore. The fifth spindle position is used for unloading and loading. These operations occupy only a part of the time available and opportunity is taken to drill the four holes in the flange before the component is removed from the chuck.

An independently operated, four spindle drill head, with a spring-loaded guide bush plate that engages the spigot diameter of the component, is introduced for drilling the holes. The drilling attachment, shown in Fig. 3, is mounted on two substantial guide bars, on which it is free to slide. Link-operated toggles on the drilling attachment engage with the pentagon centre block to provide a solid feed stroke. At a predetermined point, on the completion of drilling, the toggles disengage and the attachment is withdrawn independent of further movement of the centre block.

The drilled holes must be in a specified relationship with the special profile of the flange. Therefore, it is necessary to ensure not only that the work spindle is always stopped in a constant position, but also that the forging is always loaded in a definite position. To meet these conditions an interesting new technique is applied. Each component has a mandrel forced into the bore with a slight interference fit and a single key locating in one spline to transmit torque. Insertion and removal of mandrels are carried out on a simple hand-operated arbor press. Sufficient mandrels are available to allow the operator always to have a component mounted ready for loading into the machine.

The rear of the mandrel is in the form of a 16 deg taper to mate with the female taper of the adaptors fitted to the machine spindles. For drawing the mandrel back into the spindle, the air cylinder draw bar is fitted with a pin that engages with a bayonet slot and annulus in the rear of the mandrel. Pressure on the taper of the mandrel is sufficient to transmit adequate torque for all the operations.

A retractable sliding locator plate is used to position the mandrel, and hence the component, in correct radial relation to the spindle (see Fig. 4). It is held against the flat side of the component flange while the air cylinder mechanism is operated to draw the mandrel tightly back into the spindle. This ensures that the holes are accurately located and in correct location to the form of the component

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LUTON AUTOMOBILE DIVISION CENTRE

MR. MAURICE PLATT, M. Eng., M.I.Mech.E., is the new Chairman of the Luton A.D. Centre in succession to Mr. F. A. S. Acres. He graduated as a Bachelor of Engineering at Sheffield University in 1919 and obtained his M.Eng. in 1921. He served for some time as Technical Assistant in the experimental department of Albion Motors, Ltd.

After a period of lecturing at West Ham Technical College, Mr. Platt became technical editor of *The Motor* and held that position until he joined the engineering department of Vauxhall Motors, Ltd., in 1937, where he was until last year responsible for the design and development of passenger cars. He now holds the post of Executive Engineer.

Mr. Platt is the author of many papers and was awarded the Crompton Medal in 1929. He has been a Member of Council of the Automobile Division and formerly of the Institution of Automobile Engineers since 1937, and from 1943 to 1949 was Chairman of the respective technical committees.

LAMP OIL TRACTOR

HARRY FERGUSON, LTD., Coventry, have added a lamp oil tractor to their range. It provides a performance comparable to the existing Ferguson petrol tractor and provides 20 belt horse-power at 2,000 r.p.m. Petrol is used for starting and a three-way tap embodying a single filter and sediment bowl provides the means of transfer to the main lamp oil tank. It is not normally necessary to empty lamp oil from the bowl when restarting with petrol, since being of lower specific gravity the petrol passes over the top surface of the heavier lamp oil without mixing.

The float chamber of the Zenith carburettor has a tap so that lamp oil left in the carburettor can be drained off before restarting on petrol. The bore and stroke of the engine are 85 mm by 92 mm, and the compression ratio is 4.5 to 1. A special five-ring, solid skirt type of piston is used, having narrow type compression rings to give good sealing so that dilution is minimized. The face of the top compression ring is chromium plated. (1895)

Institution of Mechanical Engineers, Automobile Division

AIR CONDITION CONTROL

A Review of Technical Requirements and Current Automobile Practice

By C. S. Steadman, B.Sc., A.M.I.Mech.E.*

The author begins by explaining what is meant by "air condition control" as applied to vehicle interiors as distinct from "air conditioning," and proceeds to outline the technical requirements to be met. The necessity for air condition control is assessed, and a review of past practice is followed by comments on current British and American car-heating systems. The factors involved in a practical system for automobiles, including screen and window clearance, are combined in a detailed performance specification for an average British 2-litre, five-seat saloon car, while in addition, special climatic effects are considered. There are alternative sources and methods of heat and air supply, and a section is devoted to testing equipment both in the laboratory and on the road. The paper, which is concluded with a forecast of future trends of development, is supplemented by a tabulated survey of the systems employed in well-known automobile types and by a statement of the mathematical relationship between the design factors described earlier.

It is necessary to explain the phrase "air condition control" used in the title. The more common term "air conditioning," widely used in the heating and ventilating industry as applied to buildings, has a definite connotation—it implies complete control over all the factors relating to human comfort, which include freshness (oxygen and carbon dioxide content, etc.), temperature, humidity, and cleanliness of the air in an enclosed space. Unfortunately, there has been a tendency to apply the title of "air conditioning" to any device applied to a motor-car with a view to improving the comfort of its internal atmosphere; it would be equally logical to refer to the domestic open coal-fire as "air-conditioning."

The author prefers, therefore, the term "air condition control" to cover any system which improves the atmospheric comfort conditions in a vehicle, without claiming to control all the relevant factors.

Air condition control for automobiles is relatively new, particularly in British cars. However, the march of progress, assisted materially by the need for meeting export requirements, has changed this situation and already some form of heating equipment is becoming the rule, rather than the exception, on all cars produced in Great Britain. This is an overdue development, for after all, if one demands and is prepared to buy fuel to keep warm in one's home and expects heating on long distance train travel in winter, it seems logical to demand an equal degree of comfort when motoring, particularly when the necessary heating and ventilation can be obtained at relatively small capital cost and zero running cost.

It can safely be said that 90 per cent of car users in Great Britain, once they have used a car fitted with a soundly designed air condition control system for at least one winter, would insist on the provision of such equipment in any further cars they choose to purchase. In countries such as the United States, Canada, and Scandinavia, where outside temperatures of 30, 40 and even 50 deg of frost are experienced, heating is essential if Arctic clothing, frost-bite and other inconveniences are to be avoided, while screen clearance of some kind is essential if driving is to be possible.

In view of the important contribution which air condition control can make to car users' comfort and the obviously large potential market for it, the subject has been deemed worthy of considerable design investigation and development and this paper is an attempt to summarize up-to-date knowledge on the subject.

Brief review of past car heating practice

The author has little detailed information regarding the history of vehicle heating equipment. No doubt isolated attempts have been made to provide heating from the earliest days. However, prior to 1938 no British car manufacturer was supplying a heater as standard equipment on any of his models and only in the case of a few of the most expensive cars was any provision made even to facilitate the fitting of a heater. The only established heaters on the British market were hot-water heaters of the air-recirculating type, of relatively small capacity, although small-scale attempts were made at the design and manufacture of rather crude exhaust-system, fresh-air heaters. Larger recirculatory heaters were, of course, occasionally imported from the United States. Provision for heated air-feed to the windscreen on British cars was almost unknown at that time.

For many years prior to 1938 hot-water heaters of the air-recirculating type were widely available on American and Canadian cars. Most American cars had some provision for easy fitment of heaters, and for hot-air supply to the windscreen for demisting and defrosting. In 1938 only three firms of any size were making simple hot-water recirculating heaters in Great Britain.

In 1939, provision was made for the first time for built-in heating equipment by two British companies on certain of their models.

The 1939-45 war stopped all development of heating equipment for private cars in Great Britain, although some development was done in the United States during that period. When automobile manufacture was recommenced in 1945 it was seen that, in the majority of American designs, an almost complete change-over to fresh-air heating had taken place and, influenced no doubt by the necessity to provide similar equipment for export purposes, British manufacturers also showed varying degrees of interest in fresh-air heating. From 1945 until 1947 the average British car designer would only consider installing a heater of small size and hence small capacity, barely adequate for severe British winter conditions. Overseas requirements, however, have considerably changed ideas of size.

The author's company therefore re-entered the heating field in 1945 with the same heating equipment which had been standardized on one British car in 1939 and also with a small-capacity recirculating heater; meantime, however, it continued with the development of larger capacity, more elaborate, fresh-air heaters which are now coming into general use.

Heating systems in use on British and Foreign Cars

British Systems. All heating systems fitted to current British cars are of the water-heating type, that is, they derive their heat from the engine. They vary from plain, recirculatory, internal heaters to relatively elaborate fresh-air intake systems with provision for recirculation if desired.

In the case of recirculatory systems two alternative layouts are used: the first installs the heater unit inside the body (where it may be visible and audible), either on the inside face of the scuttle-pressing or under one of the front seats; car-interior air is drawn into the heater and expelled again after heating. The second layout houses the heater in a casing, located on the engine side of the scuttle-pressing, with communication between this casing and the car interior; the air recirculates by entering the casing from the car interior, passing through the heater, and then being expelled (after being heated) back into the car interior. The heater, therefore, is not visible and, if properly installed, is less audible. Moreover, this arrangement does not occupy any of the valuable interior body-space.

In the case of fresh-air systems there is some variation in the methods used. The fresh air is taken in from one of two sources, either from a forward-facing intake located behind the main radiator grille, usually at the side of the main radiator, or from the orthodox scuttle ventilator at the rear end of the bonnet. In most cases the passage of the air through the heating system is assisted by some form of electrically driven fan, although the forward motion of the car helps to produce the necessary air movement. Fans need not be in use above 30-40 m.p.h., sufficient air being supplied by "ram effect" due to vehicle motion.

With the fresh-air system, the heater unit is located either on the interior side of the scuttle-pressing (in two cases only, the author believes) or forward of the scuttle-pressing—the latter position being

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AIR CONDITION CONTROL (*Continued*)

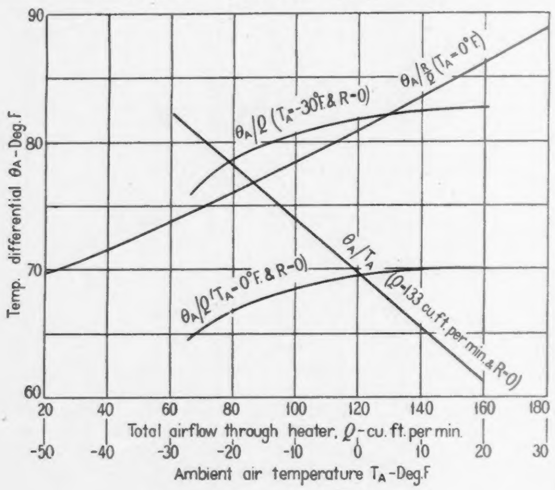


Fig. 1. Variation of temperature differential with related variables (vehicle with large-capacity heater—25 m.p.h. road speed).

Water inlet temperature to heater T_1 = 168 deg. F.; water flow, 1 gal. per min.
Q Total air flow through heater, cu. ft. per min.
R Recirculation air flow through heater, cu. ft. per min.
T_A Ambient air temperature, deg. F.
θ_A Temperature difference between mean car interior and outside ambient air (T_A), deg. F.

preferred since it economizes interior space and reduces heater noise.

Appendix I shows a classification of heater systems on British cars according to various features of the system, that is, whether fresh air or recirculatory, method and position of housing, whether fan assisted or not, position of air intake, heater capacity, etc.

Generally speaking, British heater systems are leaving the first stage of small-capacity, recirculatory heaters, barely able to cope with our own climate, and passing through an interim period of fresh-air heating with considerable variation in heater capacity.

American Systems. American heating equipment is of particular interest, as it represents the most advanced development and is designed for the most extreme conditions.

All heating systems for which design provision has been made on American cars are of the hot-water heating type and cover both recirculatory and fresh-air types. In contrast with British practice, fresh-air systems are much more general in the United States than recirculatory ones; to the best of the author's knowledge, recirculatory heating is used only in its simplest form, namely, with the heater installed inside the body on the inner face of the dash (where it is visible and audible). Because of the size of the American car, there is no difficulty in finding space even for such a bulky item as the large-capacity heaters which are in common use.

Appendix II gives a classification of the latest American heating systems for which detailed information is available, the information being grouped under the same headings as Appendix I.

The two most striking points emerging from the comparison of the British and American systems are:—

(*a*) The much larger air flows and heater output figures quoted for the American heaters.

(*b*) The fact that thermostatic control of the heating is almost universal on American systems, but is not yet used at all on British systems.

Design requirements for a practical air condition control system with recommended specification for a system on an average British five-seat saloon car

Although true air conditioning involves such additional complications as filtered fresh air, humidity control, and air cooling, in practice a satisfactory degree of comfort can be obtained in most climates by ventilation and controlled air-heating only.

Driving comfort is affected by moisture deposition on the inside, and ice formation on the outside, of the windscreen, side and rear windows, and these may best be avoided by the supply of suitably conditioned air to the glass surfaces in question, the necessary equipment becoming a part of the air condition control system.

Ventilation. Ventilation by means of window opening is difficult to control and results in draughts, and it is preferable to supply necessary fresh air through the air condition control system. From consideration of domestic, industrial and aircraft air conditioning practice, 25-30 cu ft per min per person is a suitable air supply quantity. This quantity (125 cu ft per min for a five-passenger saloon) is suitable for heating requirements and will reduce inward draughts from door and window airleaks; complete elimination of such draughts may require larger air quantities of the order of 250-500 cu ft per min, depending on how well the car is sealed.

For very hot weather, in order to provide some degree of comfort by air movement only (without actual air cooling), air quantities of 250-500 cu ft per min are necessary to be really effective and such quantities can only be obtained economically by the provision of well-designed forward-facing intake ducts with a recommended minimum total cross-sectional area of 30-40 sq in, these large air quantities being available only when the vehicle is travelling at speeds above 30 or 40 m.p.h. If such large intake ducts are provided, the fresh-air supply should be arranged to by-pass the blower and heater system altogether when heating is not required, thus minimizing resistance to flow of ventilating air.

It must be appreciated that increasing the fresh-air supply increases the amount of heating required and hence the size of the heating equipment. Space and cost considerations may limit the amount of fresh air which can be permitted under extreme heating conditions, as explained below.

Heating. Heating can most conveniently be done in a motor-car by the medium of heated air; other means such as radiant heat present considerable practical difficulties.

When the outside ambient air temperature is less than the comfort temperature required in the vehicle interior, the heat losses by conduction through body trim and skin, forced convection to outside air, and air leakage have to be met. The heat emitted by the occupants' bodies helps to a small extent in that direction.

The required heat may be supplied by recirculation heating, whereby air is drawn from the car interior through the heater and is then returned to the car interior, or by fresh-air heating, where fresh air is drawn from outside the vehicle, heated and supplied to the car interior, or by a combination of both methods. More heat has to be supplied with a fresh-air system than a recirculating one. In a typical comparison (see Appendix III) the ratio of heat requirements is 4/3.

Experience indicates that a fresh-air system is much more pleasant than a recirculatory one and down to ambient temperatures of about 0 deg F a 100 per cent fresh-air system is recommended. At much lower temperatures, however, a combination of fresh air and recirculation is preferable to minimize heater size; fresh air drawn from outside is mixed with air drawn from the vehicle interior and the mixture passes through the heater to the vehicle interior.

The standard of heating which a given system achieves on a given car is measured by the temperature differential which it maintains between the car-interior average temperature and the outside ambient temperature. This figure must state the ambient temperature to which it refers (for example, 70 deg F temperature differential with ambient air temperature of 0 deg F) because the temperature differential which a given system will maintain varies with the ambient temperature, in the case of both fresh air and recirculating systems (see Appendix IV). It is possible to calculate the relationship between the temperature differential and related variables. Figs. 1 and 2 show graphically the effect of ambient temperature and heater air-flow on temperature differential for a typical fresh-air and recirculatory system respectively.

Although a constant car-interior temperature of approximately 65 deg F may at first appear to be ideal, such an internal temperature with low external ambient temperatures of −30 to −40 deg F would result in thermal shock on entering and leaving the car, and would require an impracticably large heater. The recommended target for Canada and America, therefore, is a temperature differential of 70 deg F, with ambient temperature of 0 deg F, while for the British Isles and the milder parts of Europe, the corresponding figure is 60 deg F.

The curves in Fig. 3, based on Norwegian road tests, give an idea of the heat quantities required to give various temperature differentials at various air flows on a typical British car; the importance is seen of curtailing fresh-air supply as far as possible if heater size is to be kept reasonable, as mentioned in the section headed "Ventilation."

It must also be realized that at low road speeds and throttle openings the amount of heat available in the engine coolant is limited. Fig. 4 shows graphically figures for heat-to-water horse-power, etc., for a typical modern car. At 25 m.p.h. the heat-to-water is approximately 550 B.Th.U. per min; allowing for losses, it is useless to demand a fresh-air flow/temperature differential combination requiring greater heat output than about 450 B.Th.U. per min. It is partly for

this reason that partial fresh-air and partial recirculation heating is advocated at very low ambient temperatures.

Distribution of Heated Air in Vehicle Interior. Having arranged an adequate supply of heated air, it is important that this is suitably distributed inside the vehicle. The aim, of course, is an approximately even temperature throughout the car, with slightly higher temperatures at floor level. From the point of view of symmetry a central discharge for the heated air appears desirable, but has several disadvantages—usually there is insufficient space under the front seats to make this practicable, and unless a very large air supply is used it is difficult to project the warm air forward so as to reach the driver's feet and pedals, where cold draughts frequently occur.

Undoubtedly the centre of the scuttle or dash pressing is the ideal point from which to discharge heated air if heating for the front-seat passengers only is considered, but with most modern front-seating arrangements a single discharge at the centre of the dash will not keep the rear passengers' feet warm.

The author's experience of attempts to duct air directly to the rear seats has not been satisfactory—if the ducts are inside the vehicle body they are inconvenient and unsightly, and if outside are subject to severe heat losses; in any case space considerations usually mean that such ducts are inadequate in size and offer considerable resistance to flow.

The most satisfactory results are obtained by providing a transverse duct on the scuttle-pressing, leading warm air to each side of the car and directing the hot-air outlets from the duct so as to induce a flow of hot air near floor level toward the sides and rear of the body.

Maintenance of Screen Vision, or Demisting, Defrosting and De-icing. The duty of maintaining screen vision under cold weather conditions falls to heating and ventilating equipment and the three functions which it may have to fulfil are:—

(*a*) *Demisting,* or the prevention of moisture deposition as liquid water on the inside of the screen.

(*b*) *Defrosting,* or the prevention of moisture deposition as ice (or frost) on the *inside* of the screen.

(*c*) *De-icing,* or the prevention or removal of ice formation on the *outside* of the screen.

In demisting and defrosting, moisture deposition is caused by excess moisture added to fresh air which has entered the vehicle. This air with increased moisture content is cooled by contact with the cold screen and the excess moisture is condensed. To prevent this occurrence one of two steps can be taken. First, damp air can be prevented from coming into contact with the cold screen by blanketing its inner face with a layer of fresh air which has had no moisture added to it. This air should be projected across the screen at considerable velocity so that the damp interior air cannot mix with it until it has left the glass. Secondly, the inner face of the screen can be heated to a temperature above the dew-point of the car interior air.

With a recirculatory system, since no fresh air is available, theoretically only the latter method can be used (in practice even unheated recirculated air can be partially effective in preventing passengers' breath impinging directly on the inside of the screen), but with the fresh-air system effective demisting or defrosting can be done without actually applying heat to the fresh air. If, however, conditions are really cold, some heating of the fresh air will be necessary to prevent cold draughts in the region of the driver's and passengers' faces.

In de-icing, the solution is to raise the temperature of the outer face of the screen by means of heat applied to the inner face and, provided the heating air is distributed evenly over the inner face of the screen, then the hotter it is the more effective will be the de-icing. From the de-icing aspect there is nothing to choose between a fresh-air and recirculating system, except that it is possible to get hotter air to the screen with the latter system.

The major design problem is to provide air supply ducts and distribution nozzles of adequate size to give even air distribution over the inner face of the screen—difficulties are largely peculiar to each vehicle and experience indicates that the ideal is a discharge nozzle about 11 inches long on each side of the screen, as near as possible central with the windscreen wiper spindle, generally in accordance with Fig. 5.

Supply ducts to the nozzles should be as free from restriction as possible and ideally should not have a cross-sectional area less than approximately 2 sq in for each nozzle—this may involve two feeds to each nozzle—as in Fig. 5. Design of demister nozzles is largely a matter of "cut and try" starting from the above recommendations (see remarks on laboratory testing in Appendix VII.

In addition, it is also desirable to be able to control at will the temperature of the air fed to the screen and to be able to cut the supply off altogether at times, and certain controls are thus needed.

Controls. With a heater unit and air circulation unit of adequate capacity, whether this be a recirculating or fresh-air system, the problem of controls arises. A number of control features may be used on a heating system and these have been tabulated in Appendix V, and it is considered worth while to discuss these in detail, as follows:—

(*a*) *Car Heating Control.* If a heater of adequate capacity for very low ambient

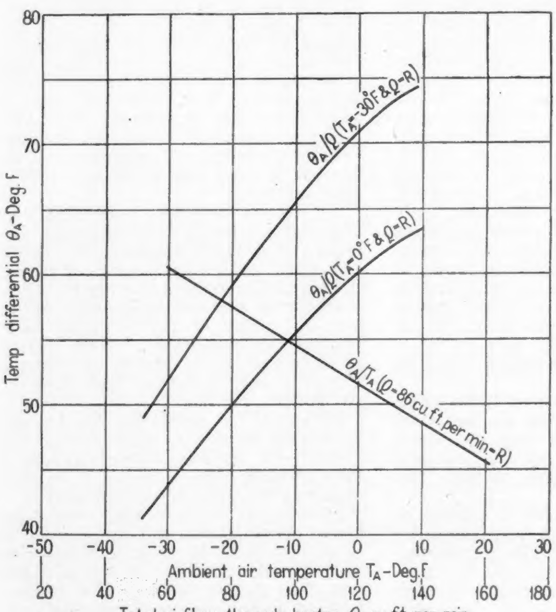


Fig. 2. Variation of temperature differential with related variables (vehicle with standard heater—25 m.p.h. road speed).

Q Total air flow through heater, cu. ft. per min.
R Recirculation air flow through heater, cu. ft. per min.
T_A Ambient air temperature, deg. F
θ_A Temperature difference between mean car interior and outside ambient air (T_A), deg. F.

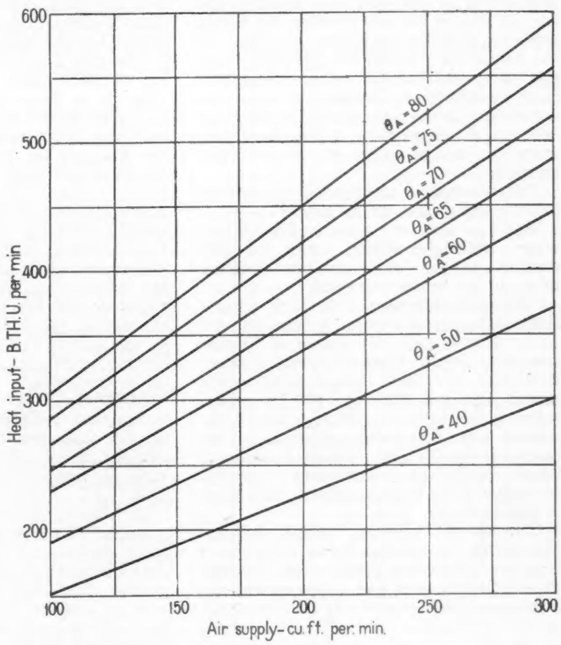


Fig. 3. Variation of heat input required with temperature differential and fresh-air supply quantity.

θ_A Temperature difference between mean car interior and outside ambient air, deg. F.

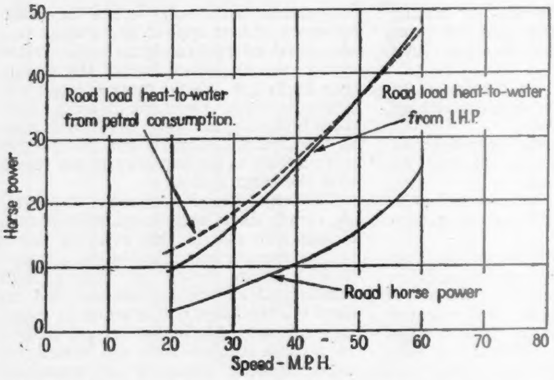


Fig. 4. Heat-to-water at road load condition for typical 2-litre car.
When calculating road horse-power, the transmission efficiency was assumed to be 95 per cent.

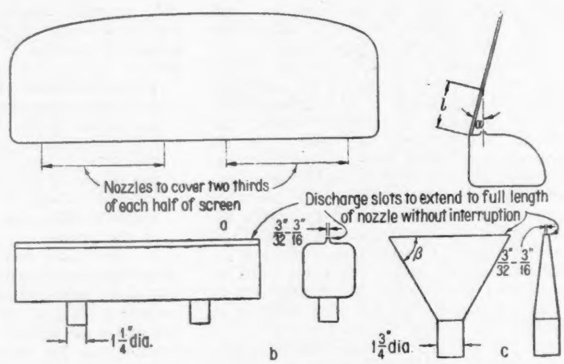


Fig. 5. Recommendations for demister nozzle design.
a The distance from the bottom of the screen of the point at which the mean line of the air jet strikes (l) should be 2-3 inches. Angle α between screen and direction of air jet from nozzle should be 30 deg.
b For a nozzle where an abrupt change of section is unavoidable, that is, where β is less than 60 deg. (see drawing *c*); two hose feeds should be used, each of $1\frac{1}{4}$-inch diameter bore.
c For a nozzle with only one hose feed, angle β should not be less than 60 deg. Also, $1\frac{3}{4}$-inch diameter bore hose should be used.

temperatures is provided, then at more moderate temperatures the heat supply will be too great. The first control requirement, therefore, is to be able to limit the heater output to any desired figure. (This requirement has not really been felt until recently on British cars, owing to the inadequate capacity of the heaters previously provided.)

A method which has been used for this purpose with only fair success on small-capacity recirculatory heaters is that of controlling the speed of the air-circulating fan by means of a rheostat. This is only really applicable to a recirculatory heater, since fresh-air systems usually have considerable airflow through the heater system due to vehicle movement "ram," even with the heater fan switched off altogether.

Even on a recirculatory heater this method is far from satisfactory if the heater is of large capacity, since although reduction of air flow through the heater does result in less nett heat input to the car the smaller air flow through the heater results in a bigger air-temperature rise. If the heated air discharge is anywhere near the occupants' feet, or other portions of their anatomy, the impact of this very hot air can be unpleasant and, in fact, may produce an *impression* of increased heat from the heater rather than a reduced amount.

Two alternative methods of controlling heater output present themselves:—

The first method varies the flow of hot water, while the second varies the proportion of the total air flow passing through the heating element.

The main difficulty of the first scheme is the difficulty of accurately adjusting the water flow to give the degree of heating required, coupled with variations of water flow due to engine speed (and hence pump pressure, etc.) and the lag which takes place between altering the water control-valve and being conscious of the change in heater output; these limitations, which apply to a manually operated control, are not so important if the control is thermostatic.

The second scheme uses a by-pass system for the ventilating air, whereby it has two alternative paths—one through the heating element and the other through a by-pass passage where it does not pick up any appreciable heat. The proportions of air flow through the heater element and the by-pass can then be adjusted to give any desired degree of heating, provided arrangements are made to mix the hot and cold air streams before they enter the car.

The latter method is considered the more satisfactory for manual operation, since the effect of a change of temperature is felt instantaneously and the actual design of a valve controlling the heated air flow and unheated by-pass air flow is a simple matter.

(*b*) *Demister-Defroster Air Heating Control.* For certain conditions in the United States and Canada it is important to supply fresh air to the inside of the windscreen with little or no heating, while maintaining full car heating; also, even in less extreme conditions, some people object to warm air being discharged near their faces—hence independent control of car heat air temperature and demister air temperature is desirable.

(*c*) *Proportioning Control for Demister and Car Heat Air.* Under average conditions demanding car heating, hot air supply will be required both to car interior for heating and to windscreen for demisting or defrosting. However, if the outside air is very dry and there are few occupants in the vehicle, hot demisting air may not be necessary and, in fact, may be objected to, so that all the hot-air supply could with advantage be devoted to heating the car interior. Conversely, under extreme frosting conditions, since driving visibility is even more important than comfort, it may prove necessary to devote the whole of the hot-air supply to the windscreen, and these requirements therefore call for a valve system capable of varying the proportions from "all air to car heating" through "part air to car heating, part air to demisting" to "all air to demisting."

(*d*) *Water Supply Shut off Valve.* This is required because unless the hot water is shut completely off from the heater element, there will always be some heat pick-up from air passing through this unit.

If a water control-valve is adopted as a means of achieving car heat control (as in (*a*) above) it is an obvious requirement for this control-valve to shut off completely at one end of its controlling range and so provide the function just called for. However, if the air by-pass method of car heat control be used, then it will be necessary to provide a water shut-off valve, but this can be linked with the air by-pass valve to avoid adding to the number of manual controls. Fig. 6 shows diagrammatically the system which the author considers ideal, together with the nearest compromise system to it in Fig. 7. The heating and ventilating sets with which the author was connected were, in both medium and large sizes, designed in accordance with the system shown in Fig. 6. Details of the manner in which these control features are achieved on modern heating and ventilating sets are shown in Fig. 8, which also shows recommended method of installation of such sets.

Specification of Five-seater Automobile Air Condition Control System. In the light of the author's experience and current practice in America and Canada, he would summarize such a specification thus:—

(*a*) Minimum fresh-air supply of 125 cu ft per min to be available from booster blower with car stationary, and to be increased by "ram" pressure from vehicle movement to about 250-300 cu ft per min at 60 m.p.h. Provision for restriction of this augmented flow under low ambient temperature, that is, heating conditions. Provision for some reasonably effective form of dust-removal equipment, to be fitted as an extra only where local conditions demand it.

(*b*) Heating equipment to be capable of maintaining a temperature differential between average inside temperature of vehicle and outside ambient temperature of 70 deg F, with outside ambient of 0 deg F. With a vehicle of the size in question and a fresh-air supply of 125 cu ft per min (allowing for restriction of "ram" effect as mentioned in (*a*) above), this would involve a heat supply of approximately 300 B.Th.U. per min.

(*c*) Means for distributing the heating air into the car so as to obtain an approximately even air temperature throughout the car body, with a tendency towards higher air temperatures at foot level.

(*d*) Fresh-air supply to windscreen to be heated or unheated at will; air quantity to be in the region of 30-40 cu ft per min total (included in figure of 125 cu ft per min given above). This air must be distributed evenly over the screen and preferably some of it should sweep all or the front part of the front side windows.

(*e*) A further important requirement is that of preventing moisture deposition on the inside and ice formation on the outside of the rear window—it is possible to reach the screen and front side windows in order

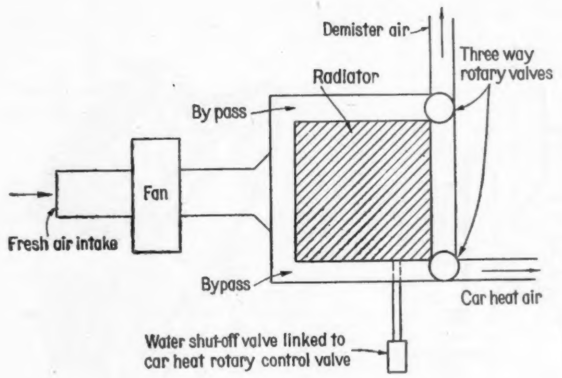


Fig. 6. Ideal heating and ventilating control system. Total—two independent manual controls.

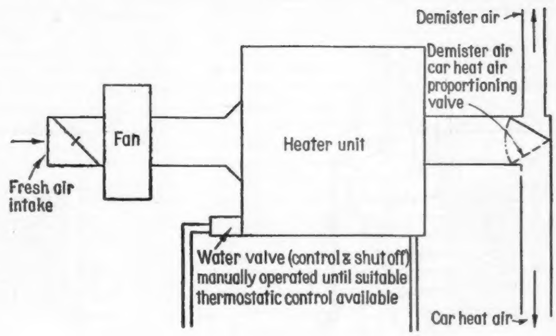


Fig. 7. Compromise heating and ventilating control system. Total—three independent manual controls.

to achieve some manual clearance, whereas the rear window is beyond the reach of the driver. However, this clearance can probably best be achieved by means of an electrically heated panel, since the cost and complication of taking heated air to the rear window are considerable.

Extra features for special climatic conditions

To approximate closer to true air conditioning under all climatic conditions it will be necessary to provide, in addition to equipment described above, the following:—

(1) Supply of clean, fresh air.

(2) Cooling, that is, refrigeration in hot weather conditions.

(3) Control of humidity in car interior.

Condition (1) implies cleaning of the fresh-air supply to the car interior, and the author as yet has no direct experience of this problem as applied to a motor vehicle. The information available on some American systems suggests that, where an air filter is fitted, it is usually discarded early in the car's life.

The basic problem is to clean the air thoroughly without the partially blocked filter offering appreciable resistance to air flow, that is, the filter will have to be very bulky or very frequently cleaned or replaced. It might be possible to develop some suitable form of washable, oil-wetted mesh filter or alternatively to achieve dust separation by centrifugal methods. Such elaborations are not essential in this country, but in some parts of the world they would represent a major advance in motoring comfort.

As regards condition (2), heat extraction necessarily involves elaborate equipment —in other words a refrigeration cycle must be used. This may be the orthodox mechanical compressor using a suitable refrigerant such as freon; or it could be the gas absorption type or the air compressor and air expansion type of equipment now being developed and used for aircraft air conditioning. All these methods involve mechanically driven equipment of some kind and at least two heat exchangers, together with automatic controls.

Appendix VI gives some calculations which indicate that the capacity needed would be in the region of at least 0.85 ton of refrigeration* or 9,900 B.Th.U. per hr minimum heat extraction.

The only information the author has regarding American practice concerns a small number of cars fitted with mechanical compression freon refrigeration; the cost was approximately £50 per car, weight increase 150-200lb, while most of the luggage space was occupied by the equipment. Currently a firm in the United States are offering equipment for this purpose with approximately 2 tons refrigerating capacity (24,000 B.Th.U. per hr heat extraction).

Condition (3), humidity control, theoretically necessitates moisture removal under some conditions and moisture addition under others. Removal, or dehumidification, can be obtained by cooling the air supply and the remarks under (2) then apply. The alternative of using a moisture-absorbing solution such as lithium chloride brine does not seem a practicable one for automobiles.

Moisture addition, or humidification, is theoretically necessary when ambient conditions are extremely cold and dry. Since the human body is quite capable of accepting relative humidity as low as 30 per cent without complaint, this effect is not serious in a typical British climate, but has a very real meaning in very cold climates.

The probable reason why humidification is never provided is that it clashes with the requirement for preventing moisture deposition on the inside of the glass surfaces, and would increase the screen-frosting tendencies unless a protective layer of dry air was provided over the screen. It has been found simpler to omit humidification and accept low relative humidity in the interests of visibility.

Summarizing, therefore, air filtration, refrigeration, and humidity control involve equipment so expensive that it can only be considered for special application for extreme climatic conditions.

Alternative sources of heat and air supply

Heat Supply. There are three practicable sources of heat supply on a motor car—the engine coolant, exhaust gases, and separate burning of a proportion of the fuel supply.

Adequate heat capacity is available from all three sources under reasonable throttle openings, but, in the case of the exhaust gases, temporary conditions when the throttle may be closed, or very nearly so, result in a shortage of available heat; this is particularly noticeable since, unlike the engine coolant, there is no heat storage capacity available.

Advantages of the engine coolant source are: adequate heat supply (except perhaps under extreme conditions), ease of control, and cheapness. Disadvantages are: necessary "plumbing" and time lag while the engine coolant warms up.

Advantages of the engine exhaust source are: speed of warm-up and higher temperatures available for defrosting purposes; disadvantages are: lack of storage capacity for heat, control difficulties, air-ducting difficulties due to heater location, and cost of suitable exhaust-gas to air heat-exchangers.

Advantages with fuel-combustion heaters are: unlimited heat supply, almost instantaneous heating available, heat available when vehicle is stationary; disadvantages are: extra consumption of fuel for heating (possibly one or two pints per hour) and heater control difficulties.

At the moment heating systems of all current British and American cars use the engine coolant as the source of heat, but it is probable that combustion heaters will be developed and introduced in the more expensive heater systems in the United States and Canada in the future.

Diesel engine limitations

The above remarks apply to heating systems for petrol-engined cars. If diesel engines should be adopted for passenger cars in the future, both engine coolant and particularly exhaust gases have a limited amount of heat available, owing to the much higher thermal efficiency of a diesel engine. This means that there is a strengthened case for fuel combustion heating. It has already been found necessary on American diesel-engined coach installations to boost the coolant heat by means of a fuel combustion heater for vehicle heating purposes, under extremely low atmospheric temperatures.

Drive for Air Supply Fans. At present the universally used method of driving heater fans is by small electrical motors. However, even with the 125 cu ft per min, which the author is postulating as a minimum for air supply, the current consumption on a 12-volt system is about 4 amps, while if a blower were used supplying 250-300 cu ft per min, which would be very desirable, one could expect current consumption in the region of 7 amps. With present electrical systems such an added load would, of course, be objectionable and the only alternative would be a mechanical drive.

Until the electrical load becomes too heavy, however, electrical drive is much more attractive than mechanical drive because of the desirability of maintaining constant or nearly constant speed for the air-supply blower. Some kind of

* A rate of heat extraction equivalent to melting one American ton (2,000lb) of ice in 24 hours.

AIR CONDITION CONTROL (*Continued*)

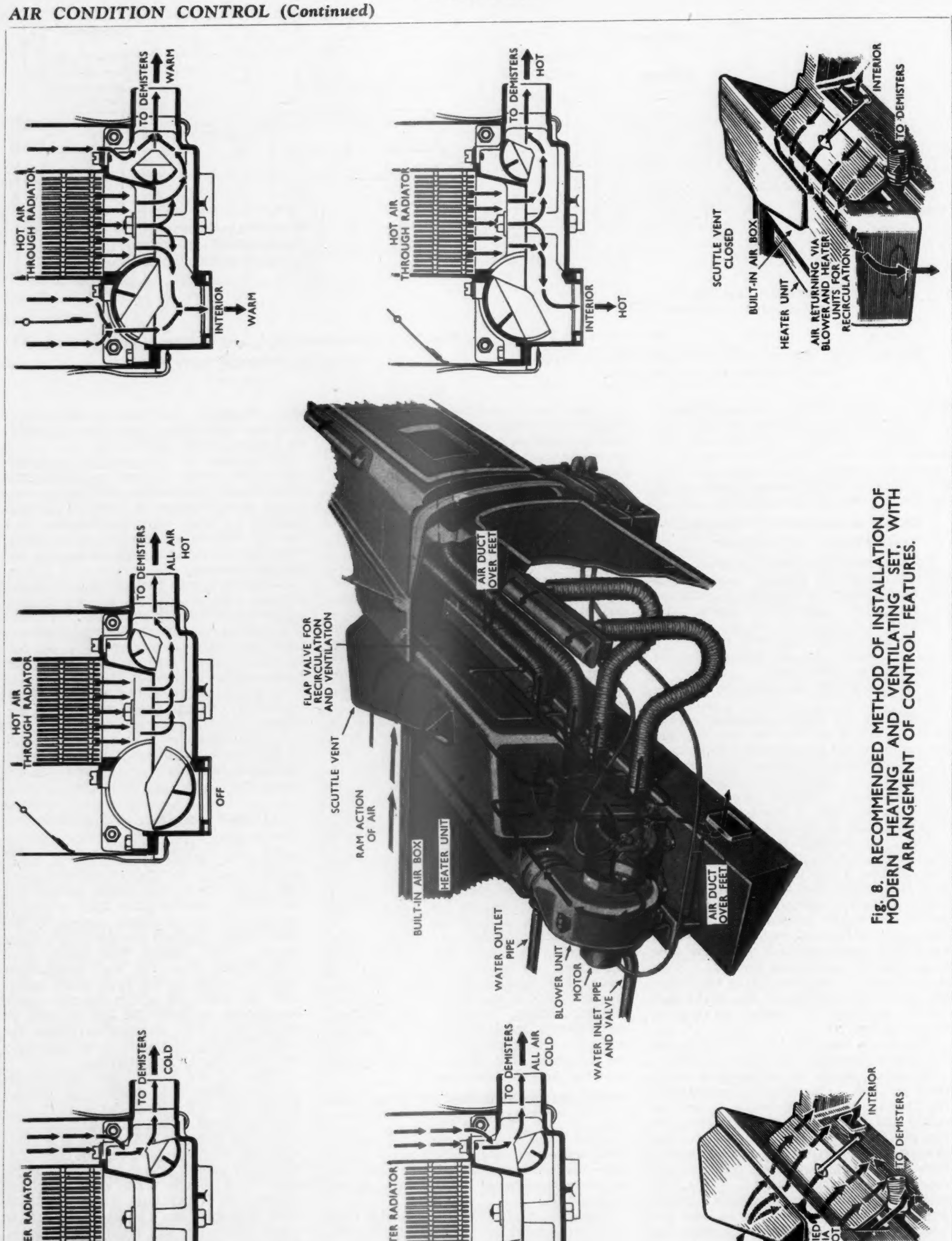


Fig. 8. RECOMMENDED METHOD OF INSTALLATION OF MODERN HEATING AND VENTILATING SET, WITH ARRANGEMENT OF CONTROL FEATURES.

mechanical constant-speed device would be required if electrical drive were not used.

Testing of air condition control equipment on the road and in the laboratory

Testing of this equipment falls into two categories—testing of the heating and ventilating units as applied to an actual vehicle installation, and testing of the same units on the bench. The latter is not too difficult and, properly conducted, can be reasonably accurate; the former, however, is a much more difficult proposition.

If really cold weather conditions exist at the time of the tests, then it is reasonably easy to evaluate the temperature differential of a given heating system and vehicle combination, but usually, of course, the weather is unsuitable when it is desired to carry out tests. Further, information regarding heater air flows, water flows, etc., is often required.

The author recently organized a test expedition to Norway in order to get some basic test data on typical British cars but unfortunately the weather experienced was not as cold as would normally be expected there at that time of the year—tests were made in conditions of 35 deg of frost and efforts were made to measure the following quantities with a reasonable degree of accuracy:—

(1) Ambient temperature.
(2) Air temperature rise across heater.
(3) Air flow through heater.
(4) Water inlet temperature to heater.
(5) Water temperature drop across heater.
(6) Water flow through heater.
(7) Air flow to each demister nozzle.
(8) Air temperatures at discharge from each demister nozzle.
(9) Air temperatures at various points (usually ten) distributed about the car interior.
(10) Relative humidity in car.

As a result of these tests, approximate figures were obtained for the heat-loss coefficients from the various vehicle bodies at the road speeds in question, while representative temperature differentials for the five vehicles tested were determined. The summarized results are given in Table 1.

Table 1. Heat-loss Coefficients and Temperature Differentials.

Vehicle	Vehicle body heat loss coefficient at 25 m.p.h., B.Th.U. per min. per deg. F. temperature differential inside to outside	Temperature differential in deg. F. maintained by vehicle heating system at 25 m.p.h., corrected to 0 deg. F. ambient temperature
A	2·1	65
B	1·4	46 (standard heater) 64 (improved heater)
C	1·3	63
D	2·0	52
E	1·4	53

As a corollary to the Norwegian road tests, tests were made on each of the vehicles in a refrigerated cold-chamber in Great Britain, in the hope of obtaining some correlation between road test results and static cold-chamber results, which would minimize in future the amount of road testing necessary. No satisfactory correlation was found on a quantitative basis as far as car heating was concerned. However, the cold-chamber tests presented a satisfactory method of evaluating quantitatively the efficiency of demisting and defrosting arrangements. A comprehensive report is available on the Norwegian car-heating test expedition.

In the case of laboratory tests, it is necessary to determine as accurately as possible the same quantities enumerated in the remarks on road tests above, together with certain others in addition.

The accurate testing of heater equipment is surprisingly difficult, mainly owing to the relatively small air and liquid flows involved, and to the sensitivity of the average heater fan to any added resistance to air flow. Appendix VII describes in some detail the methods which the author has used successfully.

Prediction of future trends of air condition control equipment

The author will only venture to predict future design trends for British vehicle heating equipment, prophecy in the case of Transatlantic progress being much more difficult. The reason is that in the major market of the United States —the home one—automobile salesmen can persuade the public of the vital necessity of any fresh piece of equipment which they can offer in advance of the next man. This fact, combined with the well-known partiality of Americans for mechanical "gadgetry," suggests that there is virtually no limit to the complexity which may ultimately be adopted in order to air condition the car interior thoroughly.

For British vehicles (not necessarily for the home market) the author's predictions are:—

(1) *On a Short-term Basis:* (*a*) Within the next twelve to eighteen months all vehicles will have provision made, as standard, for the fitment of heating equipment of adequate capacity for severe American and Canadian conditions, but equipment of smaller capacity, capable of dealing effectively with European as well as the worst British conditions, will be offered as an extra, with the large-capacity heater as an alternative.

(*b*) Heating equipment in accordance with (*a*) will be included as part of the standard equipment on at least 50 per cent of the cars built in five years from now, while the large-capacity equipment for the United States, Canada, and other extreme climates will be offered optionally at extra charge.

(*c*) Considerable improvements will be made to all classes of vehicles as regards the intake of fresh air and its distribution within the vehicle.

(*d*) Thermostatic control of heating by water thermostat sensitive to the car interior temperature, will become general on complete heating and ventilating systems.

(*e*) Equipment for dust removal from intake air will be developed considerably, and sold as an optional extra for those countries which need it.

(*f*) Efforts will be made to obtain really effective clearance on front side windows and particularly rear windows.

(2) *On a Long-term Basis:* From the point of view of long-term development much more thorough efforts will be made to design heating and ventilating equipment into new cars, right from the early drawing-board stage, particularly to obtain the benefit of large air flows by "ram" effect owing to vehicle movement instead of by an electrically driven fan, although fan assistance will be necessary at low vehicle speeds. Such designs may incorporate separate heating elements for the demisting and defrosting air, preferably housed in the actual demister nozzles or in the screen rail itself. Separate fans will almost certainly be employed for demisting and for heating and ventilating purposes, a high delivery-head fan being used for the former and a low delivery-head fan for the latter.

The author visualizes the ideal fully built-in system of the future as follows:—

(*a*) Large size scuttle ventilator, feeding fresh air to a heater radiator located beneath the scuttle ventilator—this radiator being designed for low resistance to air flow.

(*b*) Fan assistance for main heating air flow, by low delivery-pressure fan, delivery through a portion of the radiator element and then through ducts leading rearwards.

(*c*) Thermostatic control of heating for main ventilating air, by means of a thermostat sensitive to car-interior temperature.

(*d*) High delivery-head fan supplying demisting air to each nozzle; both demister fan and main ventilating-air fan driven by same electric motor, with means provided whereby motor output can be devoted entirely to demisting fan when required.

(*e*) Separate water-heated radiator elements for demister air, manually controlled by "on-off" cock in water supply line to this radiator.

(*f*) Possible addition of fresh-air intake ducts led from the front of the vehicle on each side of the car—independent of the heating system itself and only used to supply additional fresh air in hot weather.

Acknowledgments. The author would like to acknowledge the co-operation and assistance given by various motor-car manufacturers in the development of the equipment described in this paper—in particular Humber, Ltd., for their interest and support in developing heating and ventilating units, and Humber, Ltd., Standard Motor Company, Austin Motor Company, and Nuffield Organization for their collaboration in last winter's heater testing expedition in Norway.

(*Appendices begin on p. 200.*)

WINGARD (M.A.) LTD., Kingsham Road, Chichester, Sussex, have recently introduced a "totally enclosed" tyre pressure gauge. The customary open plunger that moves forward and uncovers the pressure registration figure is not employed, therefore dust and dirt cannot enter the instrument. The pressure scale is permanently shown under a perspex window. In operation, a transparent red indicator sleeve moves over the permanent scale and indicates at its top edge, the pressure in the tyre. An endwise shake or tap on the hand, throws the red marker back to zero. The standard instrument is calibrated from 0 to 50 lb per sq in, but special calibrations to any pressures, metric scales, etc., can be provided.

AIR CONDITION CONTROL (Continued)

APPENDIX I

Tabulated information for air condition control systems provided on British cars

Car (1949 model)	Fresh air (F) or recirculating (R)	Approximate heater output at 0 deg. F. ambient (car stationary), B.Th.U. per hr.	Approximate heater air circulation (car stationary), cu. ft. per min.	Type of fan: centrifugal (C) axial (A)	Position of heater in vehicle	Position of air intake to heater	Control features
1	2	3	4	5	6	7	8
Austin A.40 ..	R	4,800	59	A	Engine side of dash—starboard side	Through dash from car interior	On-off fan switch. Water shut-off valve on engine
Austin A.70 ..	R	6,200	86	A and C	Engine side of dash—starboard side	Through dash from car interior	On-off fan switch. Water shut-off valve on engine
Austin Sheerline Princess	F	5,500	41	C	Engine side of dash—central	Behind radiator grille port and starboard sides. Both ducts feeding one heater unit	On-off fan switch. Water shut-off valve on engine. Air selecting valve for demister or heated air. Air shut-off valve on intake duct
Hillman Minx	F or R at will	8,000	54	A	Casing body side of dash	Fresh—from top scuttle ventilator. Recirculating—from box below scuttle ventilator	Rheostat on fan. Door on scuttle ventilator. Recirculation flap. Water shut-off valve on engine
Humber Hawk	F or R at will	16,400	138	C	Engine side of dash—central	Scuttle ventilator	On-off fan switch. Separate air by-pass controls for car heat and demister air, giving "off," "cold," and "hot" positions in each case independently. Also, door for scuttle ventilator and recirculation dampers. Water shut-off valve on engine
Jaguar	F	8,700	59	C	Engine side of dash—port side	Behind radiator grille on port side	On-off fan switch. Water shut-off valve on heater operated from driver's seat
Jowett Javelin	R	4,800	59	A	Body side of dash—central	Radial to heater (circular)	On-off fan switch. Water shut-off valve on engine
Morris Oxford and Morris Six	R	5,500	72	A	Body side of dash—central	Radial to heater (circular)	On-off rheostat for fan. Water shut-off valve on engine
Riley ..	F	5,100	35	C	Engine side of dash	Behind radiator grille (blower fan actually forward of main radiator). Duct from fan to heater unit on port side	On-off fan switch. Air by-pass control for combined car heat and demister feed (i.e. not independent). Water shut-off valve on engine
Rover (1950 model) Large capacity Medium ,,	F	 14,500 7,600	 111 62	C	Engine side of dash—central	Behind radiator grille. Port side duct to heating system. Starboard side duct for ventilating only	On-off fan switch. Separate air by-pass controls for car heat and demister air, giving "off," "cold," and "hot" positions in each case independently. Water shut-off valve on heater coupled to car heat air by-pass controls
Singer	F	7,600	62	C	Engine side of dash—port side	Behind radiator grille	On-off rheostat for fan. Water shut-off valve on engine
Standard Vanguard (1950 model) Large capacity Medium ,,	F	 14,500 7,600	 111 62	C	Engine side of dash—central	Behind radiator grille on starboard side	On-off fan switch. Separate air by-pass controls for car heat and demister air giving "off," "cold," and "hot," positions in each case independently. Water shut-off valve on heater coupled to car heat air by-pass controls
Triumph 12T (small car)	F	7,600	62	C	Engine side of dash—central	Scuttle top ventilator	On-off fan switch. Separate air by-pass controls for car heat and demister air giving "off," "cold," and "hot" positions in each case independently. Water shut-off valve on heater coupled to car heat air by-pass controls
Wolseley 4/50 and 6/80 ..	R	4,800	59	A	Body side of dash—central	Radial to heater (circular)	On-off rheostat for fan. Water shut-off valve on engine

APPENDIX III

TYPICAL COMPARISON OF HEATER SIZE WITH RECIRCULATORY AND FRESH-AIR HEATING

The net heat output (in B.Th.U. per min. or equivalent kilowatts—1 kW. ≡ 56·4 B.Th.U. per min.) of a heater of given size is dependent on the temperature difference between the mean water temperature in the heater radiator (approximately the arithmetical mean of the inlet and outlet water temperatures) and the temperature of the air entering the heater.

The water inlet temperature to heater is kept substantially constant by the engine control thermostat and hence the mean water temperature in the heater is approximately constant—a typical figure would be about 160 deg. F.

Air enters the heater at the general car interior temperature maintained in the case of a recirculating heater (say, 60 deg. F.) and the outside ambient temperature (say, 0 deg. F.) in the case of a fresh-air system.

Consider a vehicle with body heat-loss coefficient of 2 B.Th.U. per min. per deg. F. temperature difference between body mean temperature and outside ambient air.

Consider heater circulation of 150 cu. ft. per min. (corrected to normal temperature and pressure).

Water mean temperature = 160 deg. F.

Required temperature differential between vehicle mean interior temperature and outside temperature = 70 deg. F. at 0 deg. F. ambient.

Let H_C = heater specific output required (B.Th.U. per min. per deg. F. temperature differential) for recirculating heater at 150 cu. ft. per min. (normal temperature and pressure),

and H_F = heater specific output for fresh-air heater at 150 cu. ft. per min. (normal temperature and pressure).

(1) *Recirculatory :—*

Ambient temperature = 0 deg. F.

Vehicle interior temperature = 70 deg. F.

Temperature difference, mean water to heater inlet air = 160 − 70 = 90 deg. F.

Therefore

$$2 \times 70 = H_C \times 90, \text{ or } H_C = \frac{140}{90} = 156 \text{ B.Th.U. per min.}$$

(2) *Fresh Air :—*

Ambient temperature = 0 deg. F.

Temperature difference between mean water to heater inlet air temperature = 160 − 0 = 160 deg. F.

$$(2 \times 70)+(150 \times 0·076 \times 0·24 \times 70) = H_F \times 160$$

Therefore

$$70(2+2·76) = H_F \times 160$$

or

$$H_F = \frac{333·2}{160} = 2·08 \text{ B.Th.U. per min.}$$

$$\frac{H_F}{H_C} = \frac{2·08}{1·56} = 1·33$$

In other words, the fresh-air heater has to have a specific output approximately 33 per cent greater than the recirculatory heater.

APPENDIX IV

RELATIONSHIP BETWEEN TEMPERATURE DIFFERENTIAL θ_A AND RELEVANT VARIABLES

The temperature differential which a given system will maintain varies with the amount of fresh air which is supplied to the car and with the net heater output. The net heater output in turn varies with the air flow through the heater and the temperature difference between the mean water temperature in the heater and the temperature of the air entering the heater (in practice the mean water temperature can be taken as approximately constant, since the water inlet temperature to the heater is held approximately constant by the engine thermostat).

It is because of this variation of heater output with air inlet temperature to heater that the temperature differential maintained by a given system depends on the ambient temperature. The relationship between the temperature differential which can be maintained on a given car and the relevant variables can be expressed as follows :—

$$\theta_A = \frac{H(T_I - T_A)}{(0·0182Q - 0·0182R + K)\left(\frac{1}{R_T^{0·7}} + \frac{\Delta}{2}\right) + \frac{HR}{Q}}$$

where θ_A = temperature difference between mean body interior and ambient outside, deg. F.

T_A = ambient temperature, deg. F.

T_I = water inlet temperature to heater, deg. F.

Q = total air flow through heater in cu. ft. per min. corrected to normal temperature and pressure.

R = recirculation air flow through heater in cu. ft. per min. (normal temperature and pressure).

R_T = ratio of Q to standard air flow at which heater has been bench tested.

H = specific output from heater when bench tested at standard air flow and standard water flow—B.Th.U. per min. per deg. F. temperature differential.

Δ = water temperature drop across heater per deg. F. temperature difference between water mean and inlet air.

K = mean value of heat-loss coefficient from body at road speed under consideration.—B.Th.U. per min. per deg. F. temperature differential inside to outside.

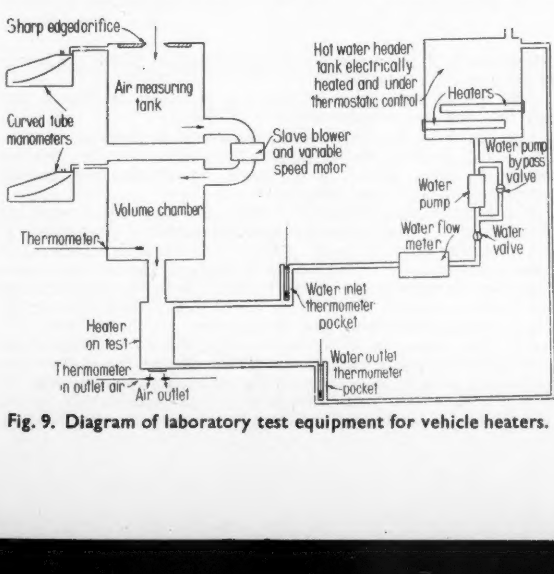


Fig. 9. Diagram of laboratory test equipment for vehicle heaters.

APPENDIX VI

CALCULATION OF REFRIGERATION LOAD FOR TYPICAL FIVE-SEAT SALOON CAR

From figures for heat transmission through the skin of the vehicle body, and also from published data on heat pick-up in hot weather with intense solar radiation in the case of coaches, heat pick-up on average five-seat car can be taken as approximately 3,000 B.Th.U. per hr. with outside conditions of 100 deg. F dry bulb and 80 deg F wet bulb, and inside conditions of 83 deg F dry bulb and 70 deg F wet bulb.

Heat leakage load	= 3,000	B.Th.U.	per hr.
Heat input from five passengers	= 2,100	„	„
Heat input as fresh air (approximately 125 cu. ft. per min. supply)	= 4,800	„	„
	9,900	„	„

Refrigeration load is 9,900 B.Th.U. per hr. or approximately 0·85 tons of refrigeration.

APPENDIX VII

DETAILS OF LABORATORY TESTING OF AIR CONDITION CONTROL EQUIPMENT

It is necessary to determine with as high a degree of accuracy as is practicable (at least within ±5 per cent) the following quantities, while the heater is operating :—

(1) Water flow through heater.
(2) Water pressure drop through heater.
(3) Water inlet temperature to heater.
(4) Water outlet temperature from heater.
(5) Air flow through heater.
(6) Air pressure (static) against which heater is operating.
(7) Air inlet temperature to heater.
(8) Average air outlet temperature from heater.
(9) Distribution of air flow from heater.
(10) Temperature distribution of air outlet from heater.

Since the source of water circulation is external to the heater, resistance to flow in the metering equipment can be overcome by the use of a suitable test water pump, and items (1), (2), (3), and (4) above are easily determined. Flow measurements can be made by both positive measurement against a time basis and by a dependable proprietary cablibrated flow-meter. Temperature measurement is made satisfactorily by glass thermometers with an accuracy of at least ±0·1 deg C installed in suitable thermometer pockets giving correct thermometer immersion and adequate turbulation and mixing of the water so as to ensure that the actual average temperatures of the inlet and exit water are accurately determined. Also, a correction must be determined and applied for heat losses from the pipes connecting the heater to the test equipment.

Air inlet and exit temperatures, items (7) and (8) above, are determined by glass thermometers with accuracy of at least ±0·25 deg C, and care must be taken to ensure that the real

AIR CONDITION CONTROL (Continued)

APPENDIX II

Tabulated information for air condition control systems provided on American cars

Car (1949 model)	Fresh air (F) or re-circulating (R)	Approximate heater output at 0 deg. F. ambient (car stationary), B. Th.U. per hr.	Approximate heater air circulation (car stationary unless otherwise stated), cu. ft. per min.	Type of fan: centrifugal (C) axial (A)	Position of heater in vehicle	Position of air intake to heater	Control features on heater
1	2	3	4	5	6	7	8
Buick	F	24,000	160 (at 30 m.p.h.)	1–C	Under seat passenger side	Behind radiator grille	Thermostatic water valve
Cadillac	1–F and 2–R	12,000 (F) 7,500 each (R) 27,000 total	0 m.p.h.: 120 (F) 140 each (R)	1–C (F)	Fresh-air unit on dash	Behind radiator grille (fresh-air unit only)	Thermostatic water valve on fresh-air unit
		22,000 (F) 7,500 each (R) 37,000 total	60 m.p.h.: 180 (F) 140 each (R)	2–A	Recircualating units under seats		Switches on recirculator fans
Chevrolet "Airflow"	F	15,000	130	—	Engine side of dash	Behind radiator grille	Thermostatic water valve
Chevrolet	R	10,000	120	—	Body side of dash	—	Fan switch
Chrysler 550 ..	F	22,000	130	2–C	Engine side of dash	Behind radiator grille, but not facing forward	Thermostatic water valve
Dodge 300 ..	F	20,000	130	1–C	Engine side of dash	Behind radiator grille, but not facing forward	Louvres and fan switch
Ford	R	10,000	120	—	Body side of dash	—	Fan switch
Ford, Mercury, Lincoln "Magic Air"	F	22,100	150	—	Body side of dash	Behind radiator grille	Thermostatic water valve
Hudson	F	10,000	130	—	Body side of dash	Hood cowl vents	Thermostatic water valve
Kaiser-Frazer ..	F	10,000	150	—	Body side of dash	Behind radiator grille	Thermostatic water valve
Kaiser-Frazer Special	R	15,000	180	—	Body side of dash	—	Fan switch
Kaiser-Frazer De Luxe	F and R	15,000	200	—	Body side of dash	Behind radiator grille	Fan switch
Nash "Weather Eye"	F	18,000	—	1–C 1–A	Body side of dash	Hood cowl vent	Thermostatic water valve
Oldsmobile.. ..	F	16,500	130 (at 25 m.p.h.)	—	Under driver's seat	Behind radiator grille	Thermostatic water valve
Packard	F	10,000	150	—	Body side of dash	Behind radiator grille	Thermostatic water valve
Plymouth 100 ..	R	10,000	120	—	Body side of dash	—	Fan switch
Pontiac	F	16,500	130 (at 25 m.p.h.)	1–C	Under driver's seat	Behind radiator grille	Thermostatic water valve
Studebaker Climitizer	F	20,000	130	2–C	Under passenger's seat	Side cowl vents	Manual water valve

APPENDIX VII (continued)

average of the inlet and of the exit air temperatures is determined, since temperature distribution throughout the inlet or exit air may vary considerably; multiple thermometer readings are used to arrive at the average and also to give a picture of the temperature distribution itemized under (10) above.

Distribution of air flow from heater can be explored where necessary by means of the proprietary "Velometer" which is very suitable for this purpose; alternatively an orthodox small pitot head can be used.

The two remaining items, (5) and (6), are the most difficult to measure accurately and they are best determined by means of a special arrangement of test apparatus as described below. The problem is that, in order to measure air flow accurately, the metering device must in itself result in an appreciable air pressure drop and with fans of the type normally employed in vehicle heating equipment, such additional air pressure resistance in itself affects the fan performance. Further, unless smooth air flow conditions apply at the inlet and exit to the heater, accurate measurement of the actual air resistance against which the heater operates is very difficult.

The special equipment in question is shown in Fig. 9 and may be described as follows:—

The heater to be tested is connected to a plenum chamber from which it draws its air supply, the air flow into the heating equipment being exactly as it would be entering the heating equipment on an actual installation.

The plenum chamber itself is fed with air from a metering chamber, via a variable-speed centrifugal blower of large capacity. The metering chamber itself is fitted with multiple sharp-edged orifices which can be plugged up at will. Pressure tappings to determine accurately the pressure in the metering chamber and the plenum chamber are used in accordance with B.S.S. 1042—1943 recommendations, and the design of the sharp-edged orifices should also be in accordance with the same specification.

All the air passing through the heater on test passes from the test room atmosphere, through the metering orifices into the metering chamber, then through the centrifugal booster blower into the plenum chamber, and from the plenum chamber through

APPENDIX V

Table of control features which may be used on a vehicle heating system

Name of control	Function	Classification	Alternative methods of control for this purpose	Recommended methods	Remarks
(1) Car heating control	To control amount of heating	Essential	(*a*) Fan motor rheostat (recirculating systems only) (*b*) Fan motor rheostat or on-off switch and throttle valve in fresh air supply duct—fresh air systems only (*c*) Water throttle valve (manually operated) on water supply line to heater—fresh or recirculator system (*d*) Water throttle valve (thermostatically operated but with manual datum setting) on water supply line to heater—fresh or recirculator system (*e*) Air by-pass on heater—fresh or recirculator system	(*d*) or (*e*)	
(2) Shut-off valve on air supply to car interior (applies to fresh air systems only)	(*a*) To prevent entry of cold fresh air into car interior before engine circulating water has warmed up (*b*) To exclude dust, fumes, etc., when necessary	Essential	(*a*) Shut-off valve in fresh air supply duct to heater (*b*) Separate shut-off valves in ducts taking air from heater to car interior and windscreen	(*b*) (see also 4(*b*) below)	
(3) Demister and defroster air heating control	To control temperature of air fed to screen	Desirable	(*a*) Separate air by-pass valve on heater for demister air supply—fresh or recirculatory (*b*) Independent radiator element for demister air with its own manual water throttle valve	(*a*) or (*b*)	It is sometimes required to supply cold air to screen while hot air is supplied to car interior NOTE.—Under some very cold Canadian and United States conditions, dry powdered snow is cleared by wipers satisfactorily provided no heat is applied to screen—however, inside of screen must be curtained with fresh cold air to prevent internal frosting
(4) Proportioning control for demister and car heat air	To enable proportion of total air supplied to be varied as between demister air and car heat air. Also, to enable all air to be fed to screen when necessary	Desirable	(*a*) Two-way valve which in one extreme position supplies all air to car interior and cuts off demister air supply and, in other extreme position, *vice versa* (*b*) Separate shut-off valve in demister air duct—see 2(*b*) above	(*a*) or (*b*)	
(5) Water supply shut-off valve	To prevent any heat input whatever to car, even by radiation from heater in hot summer conditions	Essential	Water shut-off valve	Water shut-off valve combined with water control valve where thermostatic heater control is used (see (1) above) or Water shut-off valve linked with air by-pass valve where manual control of heating is used (see (1) above)	

APPENDIX VII (continued)

the heating equipment on test back to the test-room atmosphere. Since the heating equipment is then discharging against atmospheric pressure, the pressure difference across it is represented by the difference in pressure between the interior of the plenum chamber and the atmosphere. When the plenum chamber pressure is equal to atmospheric the heater is operating with free air entry and exit. When it is below atmospheric the heater is operating against a resistance air pressure, and when it is above atmospheric, the heater is operating with a "rammed" air supercharged intake.

In an actual test, sufficient metering orifices are opened to allow the required air flow with a pressure drop across the orifices of about 2-4 inches of water, and the centrifugal booster blower speed is adjusted to give a negative pressure in the plenum chamber corresponding to the air pressure resistance against which the heater is to be tested. The air flow can then be determined from the air pressure reading in the metering chamber and the calibration curve of the orifices in use. Measurements of air and water temperatures, water flows, etc., are made under these steady conditions.

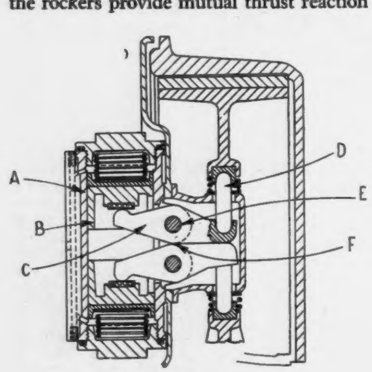
Testing of Demisting and Defrosting Equipment. The easiest method of evaluating approximately and qualitatively the probable screen distribution is to "mock-up" the nozzles in correct relationship to a surface wetted with petrol or other volatile liquid (representing the screen). By running the demister air supply the evaporation patterns on the surface will give a good indication of the air distribution. To prove a system out quantitatively, however, the best method, in the absence of suitable weather conditions, of course, is a static cold-chamber test (see account of similar tests in report on Norwegian and associated tests), which will give a definite indication as to whether the screen clearance system will be satisfactory under severe conditions.

CURRENT PATENTS

A Comprehensive Review of Recent Automobile Specifications

Externally mounted brake cylinder

BY mounting the actuation cylinder on the outer side of the brake plate it is exposed to the air stream and relieved of heat generated in the drum. Cylinder A contains two spring-loaded pistons B bearing respectively on ends of a pair of rocking levers C which, through push rods D, transmit effort to the brake shoes. Although rockers C are located on and pivot about pins E, the latter are not subjected to thrust. Arcuate contours F on the rockers provide mutual thrust reaction



No. 614853

faces, and in an alternative embodiment could be constituted by ball races.

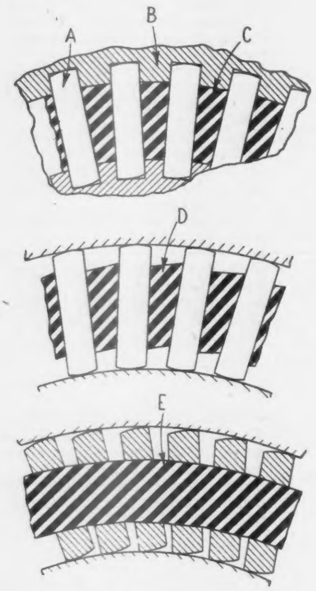
The system is completely balanced; there being equal and opposite force both in the cylinder and applied to the shoes. Pivot pins are relieved of thrust and consequently there is no reaction communicated to the brake plate. The whole of the mechanism can be enclosed and lubricated and, it is claimed, is easily produced and assembled. *Patent No.* 614853. *Régie Nationale des Usines Renault (France).*

Overrunning clutch

IN one-way clutches of the sprag type it is necessary to provide a resilient coupling between the sprags to bias them into engagement with the cylindrical races in order to ensure the clutch will engage instantaneously when torque is transmitted in one direction of rotation. In this invention the resilient member is formed of a synthetic rubber, such as "Neoprene" which is not affected by oil and may therefore be used in connection with the lubricated elements.

Sprags A of conventional form are located radially in a moulding fixture B and wedge-shaped spacers C of synthetic rubber are vulcanized to the sprags to form an annulus. In order to insert this annulus between the clutch races it is necessary that the sprags be inclined from their normal radial position. This will produce a slight deformation of the spacers C, as shown at D, and this in turn creates the necessary bias towards the original position.

In a modified construction the sprags are held in spaced relation to each other by an annular member E of synthetic rubber. This engages grooves or chan-



No. 615670

nels in the sprags and is, of course, vulcanized to maintain location.

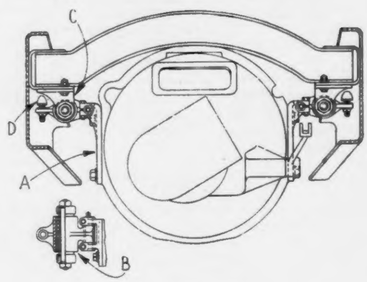
In another method the sprags may be assembled in a continuous strip of "Neoprene" which may be cut into sections of appropriate length and inserted between the co-operating clutch surfaces. Thus the same strip may be used for forming units applicable to clutches of different radial dimensions. *Patent No.* 615670. *The Gear Grinding Machine Co. (U.S.A.).*

Engine suspension

THIS invention is concerned with the lateral supports of a three-point engine suspension system adapted to absorb vibrations and resist torque reaction. Each side mounting comprises a bracket A bolted to the bell housing and carrying a fulcrum pin which passes through a bearing bush in one end of the lever B. This lever is mounted on a spindle carried by bracket C secured to a frame cross member, with an interposed torsionally resistant rubber bushing. The outer end of the lever is provided with an adjustable rubber buffer D.

Any downward movement of the engine tends to draw the pivotal centres of the levers closer together, thus exerting a compressive action on the rubber bushings which, in conjunction with the normal shearing action, has the desired effect of absorbing vibrations and torque reaction forces. The axes of the lever fulcrum pins may be either parallel to the axis of the crankshaft or parallel to the longitudinal axis of gyration of the engine.

In an alternative system one of the levers may be replaced by a rubber-



No. 617552

bushed pivotal connection between the engine and the frame. *Patent No.* 617552. *E.R.F., Ltd., and E. Sherratt.*

Body construction

TO carry the wheel mountings and sustain the main transverse and vertical loads from the road wheels, vertical bulkheads are arranged approximately over the axes of the wheels. They may be essential parts of a chassisless construction or be attached to longitudinal side members to form virtually a chassis on which the remainder of the body is erected. In the example, front bulkhead A and rear bulkhead B are secured to side members C. Independent wheel suspension units D are carried from the front bulkhead. To support the engine a forward extension E is braced to the bulkhead by members F. *Patent No.* 613784. *Rover Co., Ltd., and M. F. C. Wilks.*

No. 613784